COMBUSTION

20 1936

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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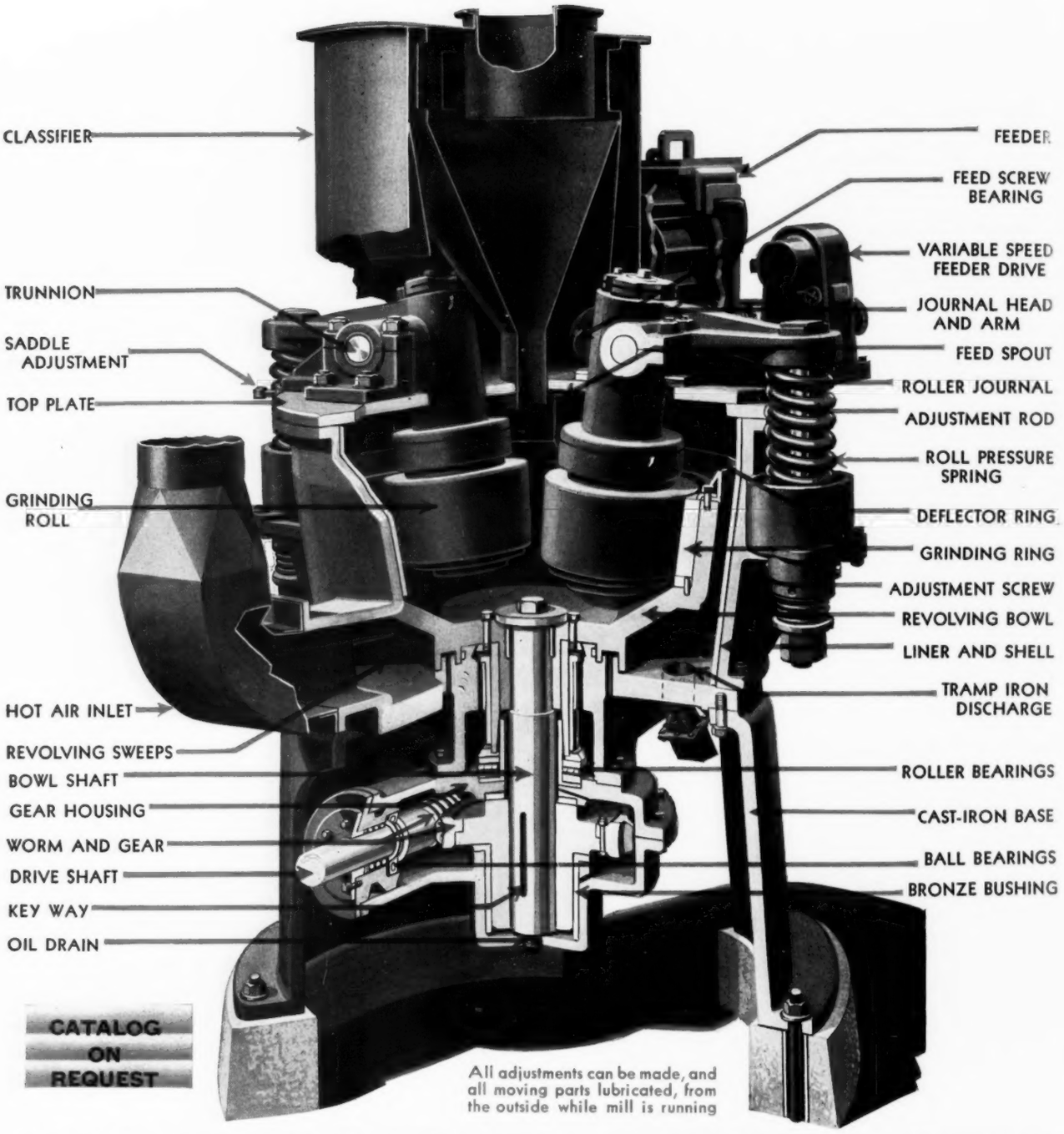
Interior of Conners Creek boiler drum showing feed trough and supply pipe

Effect of Type of Surface on Radiation Loss from Furnaces

James Watt, Engineer and Scientist



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COMBUSTION ENGINEERING

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME SEVEN NUMBER SEVEN

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EDITORIAL

Caustic Embrittlement

Many will recall the controversy that waged some years ago following an epidemic of boiler metal cracking which was attributed to so-called "caustic embrittlement." The extensive researches by Professor Straub appeared to fix the blame definitely on sodium hydroxide and, although at variance with certain German investigators, the conclusions were quite generally accepted. Subsequent studies led to establishment of the sulphate-carbonate ratio by the A.S.M.E. Boiler Code Committee which has proved effective in inhibiting embrittlement.

With a view to establishing further data as to the effect of various solutions on deterioration of boiler metal, and particularly the most desirable sulphate-carbonate ratios in the higher pressure and temperature ranges, an extensive series of investigations was undertaken at the New Brunswick, N. J. Station of the U. S. Bureau of Mines. Much valuable data have been obtained and nine progress reports have been submitted.

The most recent development in the study of so-called embrittlement is the discovery of the effect of silica. In attempting to confirm the results of Professor Straub, the investigators at New Brunswick repeated the former's experiments, using chemically pure caustic soda, and were unable to produce cracking of the boiler metal. This led to further investigation which disclosed that Professor Straub had used a commercial grade of caustic that contained a small amount of impurities. The effect of each of these was then studied and it was found that silica was the principal constituent. That this conclusion has been affirmed by Professor Straub in some recent experiments, was brought out in the discussion at the A.S.M.E. Annual Meeting in December.

This revelation means much to the investigators and may ultimately have important bearing on boiler water treatment, but it is likely to have little immediate effect on boiler operation. It is commercial rather than chemically pure caustic with which the boiler operator has to deal. However, the increasing use of welded drums and the recommended sulphate-carbonate ratio provide a means of successfully combatting its effects.

Our Petroleum Reserves

Periodically during the past twenty-five years there have been warnings of approaching exhaustion of our domestic petroleum reserves, based on apparently authoritative estimates of the oil remaining in the ground. Each time, however, new pools, improved technique in drilling and increased recovery from existing wells have brought unexpected supplies and tended to discredit the professional forecasts. Moreover, development of the cracking process to produce large yields of gasoline from the heavier oils has made it possible to keep up with the ever-increasing demands for motor fuel.

This situation has been responsible for a widespread popular belief that sufficient oil would be available almost indefinitely, despite the fact that only two per cent of the wells produce fifty per cent of the oil. The fallacy of this assumption is discussed by Messrs. Snider and Brooks in a treatise recently issued by The Chemical Foundation, which is well worth reading by those concerned with the use of fuel.

The authors assert that, based on the latest geological knowledge, our reserve is equivalent to twelve years' consumption at the present rate, but that there is no method now known by which all this oil can be brought to the surface in twelve years; also, that a shortage of domestic production, necessitating increased imports and increased prices, may be expected within five to eight years, even though actual exhaustion may be many years off. Inasmuch as imports are uncertain in periods of emergency, and as adequate domestic reserves must always be maintained to meet the needs of national defense, it is suggested that now may be time to look to the future.

Among the possible substitutes are oil from shale, of which there is abundance in this country, the hydrogenation of coal, as is now being practiced abroad, and perhaps a combination of hydrogenation and pressure cracking. However, the price ranges of petroleum in the United States have not been high enough to inspire the production of such substitutes on a commercial scale. When a shortage becomes imminent the increasing prices will undoubtedly warrant their production. It is pointed out that much development work and considerable time will be required before such an industry can be operative on a large commercial scale.

Record Central Station Output

Despite the enactment of restrictive legislation, governmental competition and local hinderances, the electric utilities established an all-time record in power generation for the year just closed. Preliminary figures of the Edison Electric Institute place the electricity generated as approximately $93\frac{1}{2}$ billion kilowatt-hours which represents an increase of about ten per cent over that of 1934 and four per cent over the banner year 1929. This figure is exclusive of electricity generated for public use by electric railways.

While the large commercial load was not back to the 1929 level it showed a most encouraging increase of more than eleven per cent over 1934. On the other hand, the domestic load for 1935 exceeded that of 1929 by more than forty per cent although the gain over 1934 was not so marked.

When it is considered that the very considerable increase in load during the Fall of last year has been sustained, the outlook for the present year is most encouraging.

Modernizing the Conners Creek Power Plant—III

Boilers and Steam Generating Equipment

This is the third of the series of six articles describing in detail the rebuilding of the Conners Creek Power Plant of the Detroit Edison Company. The introductory article in the November issue dealt with the economic considerations involved; that in the December issue described the stokers and development of their combustion air control; the present article deals with the steam-generating equipment; and that in the February issue will describe the plant operating cycle and heat balance. It will discuss how advances in the art of power generation have been incorporated in the rebuilt plant; comparison will be made of the new and the old auxiliary power system and drives; and a brief account will be given of how portions of both the old and new equipment were operated simultaneously during the rebuilding program.

By SABIN CROCKER
Engineer, The Detroit Edison Company

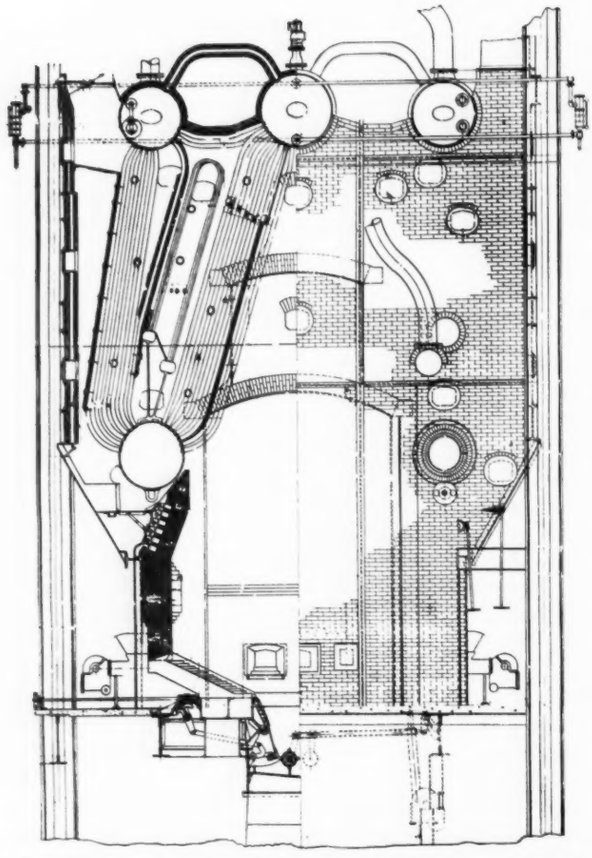


Fig. 18—Cross-sectional elevation of old Conners Creek boiler installed in 1915

THE notable advance in heat-power engineering made since Conners Creek originally went into service in 1915 is particularly evident when comparing the new with the old steam-generating equipment. A marked contrast between the 1915 and the 1935 Conners Creek boilers, both within the same column spacing, is shown in Figs. 18 and 19.

Trebling the kilowatt equivalent of the boiler capacity within the existing column spacing as has been done in this case, resulted from the more efficient heat cycle in conjunction with the following modernizations in boiler design: (1) Increasing the continuous rate of coal burned per square foot of stoker grate area to permit a high rate of steam generation for long periods; (2) adopting a higher steam pressure and temperature as permitted by modern materials and methods of construction; (3) maintaining a much higher rate of heat liberation in the furnace which has been made possible by the use of air heaters, water-cooled furnace walls, screen tubes shielding the radiant portions of the superheater and higher velocities of gases carrying heat to other parts of the boiler; (4) adding welded jointless economizers which impart to the feedwater about seven per cent of all the heat added in the boiler room; (5) maintaining uniform superheated-steam temperature by combining radiant surface on both sides of the boiler with convection surface installed on one side only and arranged so that the gas flow can be split between two sides of the boiler so as to sweep through the convection surface as required to give essentially the same temperature at all ratings; and (6) developing an improved design of soot-blower elements which are more effective in keeping the tube surface clean of fly ash at continued high ratings and which, to a limited extent, are capable of descaling fused slag from tubes facing the furnace without having to drop the load to produce this result.

As the stoker and air control have been fully described in the preceding section on "Stokers and Combustion Air Control,"[1] the present article is confined to the remaining features of the above itemization, together with other points of interest.

Heat Released and Transferred

How the effective kilowatt capacity of each boiler-room building-column space was increased by the design of the new boilers and the conditions of operation can

[1] COMBUSTION, December 1935.

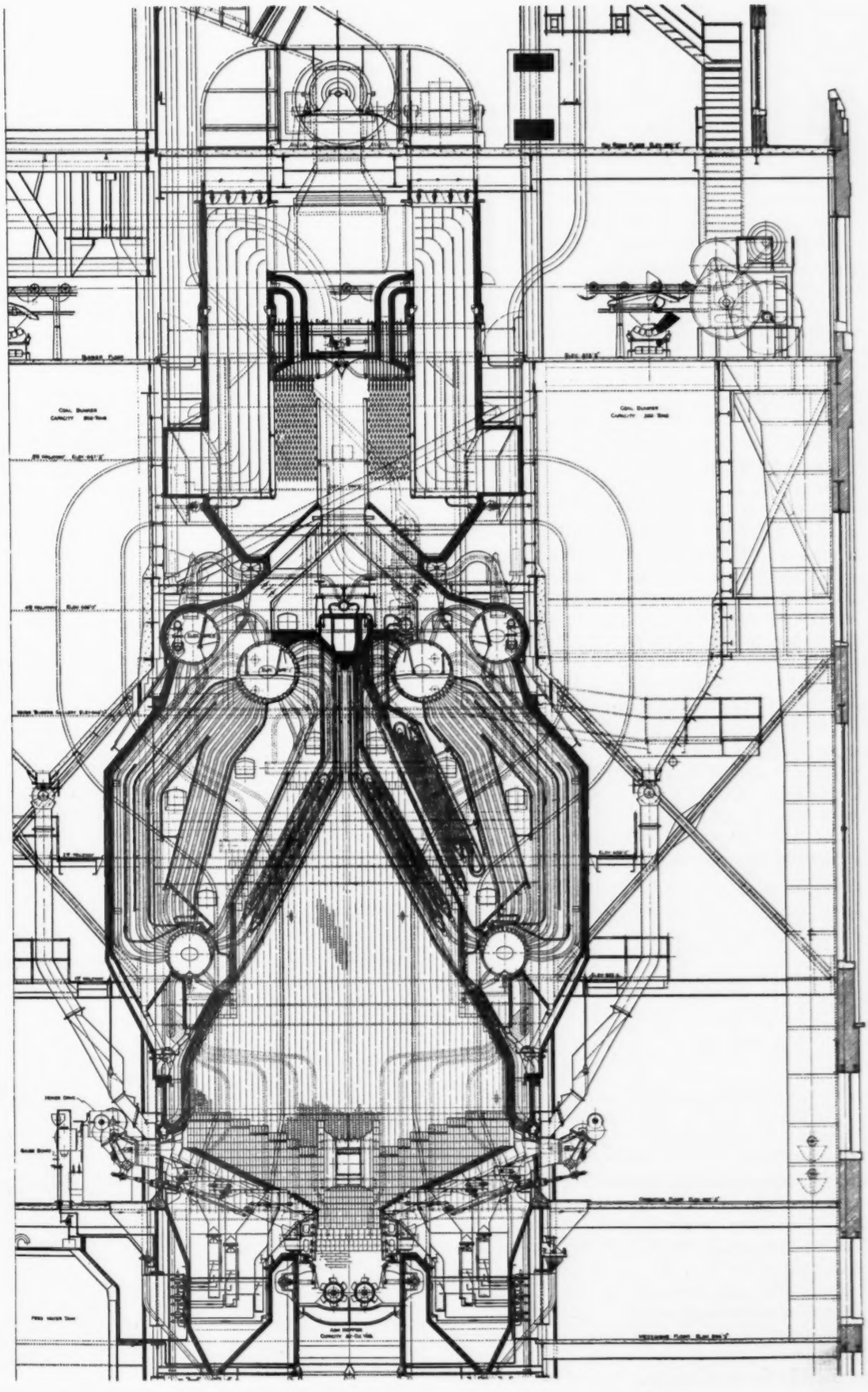


Fig. 19—Cross-sectional elevation of new Conners Creek boiler

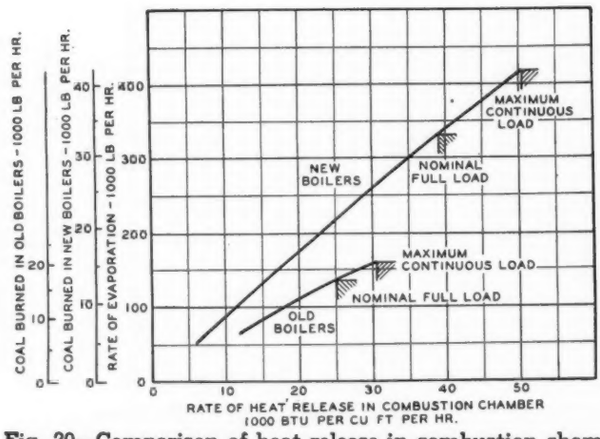

COAL BURNED IN OLD BOILERS - 1000 LB PER HR.
COAL BURNED IN NEW BOILERS - 1000 LB PER HR.
RATE OF EVAPORATION - 1000 LB PER HR.
MAXIMUM CONTINUOUS LOAD
NOMINAL FULL LOAD
NEW BOILERS
MAXIMUM CONTINUOUS LOAD
NOMINAL FULL LOAD
OLD BOILERS
RATE OF HEAT RELEASE IN COMBUSTION CHAMBER
1000 BTU PER CU FT PER HR.

Fig. 20—Comparison of heat release in combustion chamber, new vs old Conners Creek boilers

best be demonstrated by a brief exposition comparing the heat transfer rates in the various sections of the boiler equipment.

The rates of heat release in the combustion chambers are compared in Fig. 20. The old boiler with 8705 cu ft of furnace volume had a rate of heat release up to 30,500 Btu per cu ft per hr against 50,400 Btu or better for the new boiler with a 11,000 cu ft furnace volume.

A typical maximum continuous output of the old boilers is the 159,805 lb per hr steam rate at 224 lb per sq in. gage and 634 F of test No. 9 as reported in the A.S.M.E. technical paper, "Tests of a Type-W Stirling Boiler at Conners Creek Power House" by P. W. Thompson.[2] This indicated 190,970,000 Btu per hr imparted to steam generation. Taking this heat-transfer rate for the old boilers as a 100 per cent datum, the heat-output capacity of the new equipment has been raised to about 248 per cent by the percentage of heat transferred by the agencies indicated in Table VI. This takes into consideration only the heat added above the incoming boiler feedwater temperature. A comparison based on what might be termed the normal rating would show approximately the same ratio.

TABLE VI

COMPARISON OF THE HEAT IMPARTED ABOVE THE FEEDWATER TEMPERATURE AT MAXIMUM RATING OF THE NEW VS THE OLD STEAM-GENERATING EQUIPMENT OF THE CONNERS CREEK PLANT

Heat Transfer Agent	Old Plant, Per Cent	New Plant, Per Cent	New Plant as Per Cent of Old Plant Total
Economizers		6.7	16.8
Boiler	88.7	72.7	180.4
Superheater	11.3	20.6	51.2
Total	100.00	100.0	248.4

The higher rate of heat release in the combustion chambers of the new boilers was made possible by the use of preheated air and by completely surrounding the fire with water-cooled walls and superheated-screen tubes. Estimating roughly, approximately 40 per cent of the total evaporation takes place in these tubes surrounding the furnace. Test data, which it is hoped to obtain at a later date, are expected to confirm this estimate or even show a still higher proportion. A view of the combustion chamber, water-wall and superheated screen tubes is shown in Fig. 21 which is self-explanatory.

The figures given in Table VII on heat-transfer surface in the steam-generating equipment complete the account of heat released in the new boilers.

[2] *Transactions* A.S.M.E., 1922, Vol. 44, p. 1005.

TABLE VII

SHOWING THE DISTRIBUTION OF HEAT-TRANSFER SURFACE IN THE NEW CONNERS CREEK STEAM-GENERATING EQUIPMENT

Type of Surface	Sq Ft of Effective Heating Surface		
Water Walls			
Fin tubes			
Exposed halves of tubes	1,557		
Exposed fin surface	4,017		
Total fin tube surface		5,574	
Integral block		303	
Total water-wall surface			5,877
Boiler, saturated surface			26,764
Superheater			
Radiant surface at radiant side	1,207		
At radiant-convection side	1,655		
Total radiant surface		2,862	
Total convection surface		5,487	
Total superheater surface			8,349
Economizers, two single-pass, combined			
Exposed tube surface	3,687		
Exposed fin surface	4,933		
Total economizer surface			8,620
Air heaters, two-plate type, combined			28,980

Electrically-welded Boiler Drums

The use of electric-fusion-welded boiler drums for the first time in this Company's experience is in keeping with the decision to have a throttle pressure of 600-lb gage in a modern regenerative feedwater-heating cycle of a relatively high Carnot efficiency. The *W* or *VV* type of boiler, having drums some 54 in. in diameter, offers several problems when contemplated for pressures in excess of 400 lb. The relatively large drum diameter requires the use of walls up to 4 in. thick to meet the strength requirement in the ligaments between tube holes.

Since riveted joints cannot be efficiently designed for plate thickness in excess of 3 in. and pressures of the order of 600 lb, it becomes necessary to use either forged drums made from solid billets or else to use rolled plates with welded seams. The almost prohibitive cost of forged

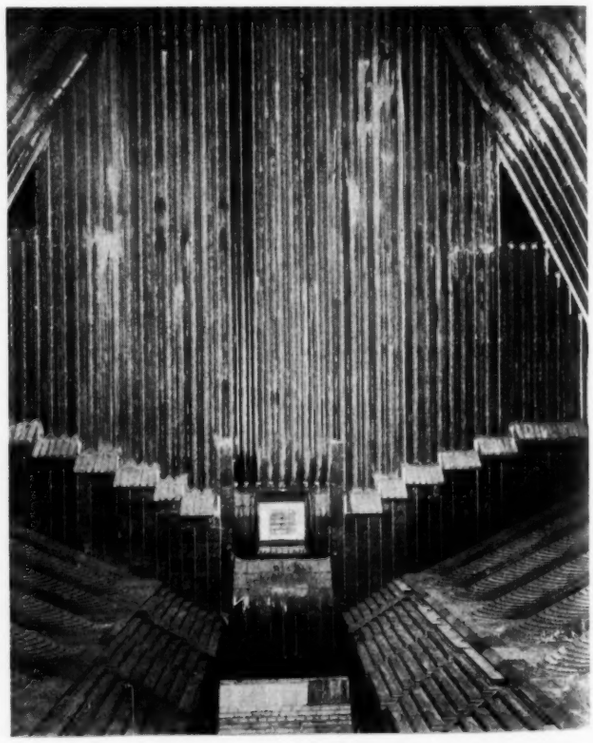


Fig. 21—View of combustion chamber of new Conners Creek boiler

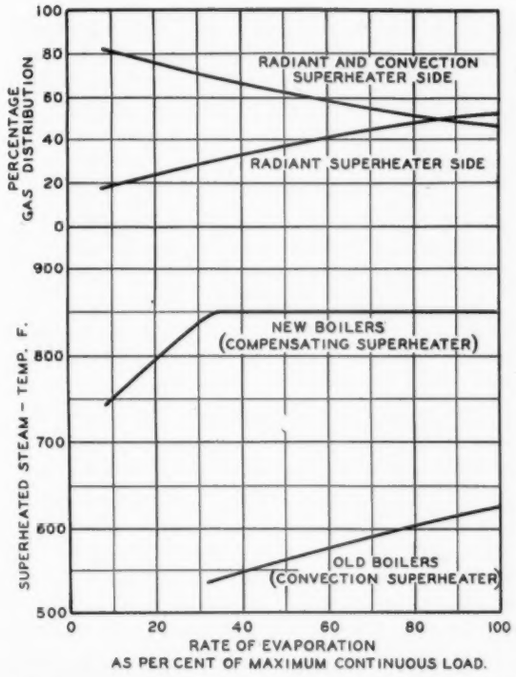

PERCENTAGE GAS DISTRIBUTION
RADIANT AND CONVECTION SUPERHEATER SIDE
RADIANT SUPERHEATER SIDE
SUPERHEATED STEAM - TEMP. F.
NEW BOILERS (COMPENSATING SUPERHEATER)
OLD BOILERS (CONVECTION SUPERHEATER)
RATE OF EVAPORATION AS PER CENT OF MAXIMUM CONTINUOUS LOAD.

Fig. 22—Comparison of steam temperature in new and old Conners Creek boilers

drums of the size and wall thickness required for large bent-tube boilers practically restricted the use of this type of boiler to pressures within the limits of riveted-drum construction until the manufacture of electric-fusion-welded drums was permitted under the A.S.M.E. Boiler Code.

The adoption of fusion welding for the fabrication of these boiler drums introduced several new possibilities in design. Perhaps the outstanding contribution was the ability to make the sector containing the tube holes out of substantially thicker plate than the remainder of the drum which had to withstand only the ordinary bursting stresses. In this method of construction the heavy plate which is to contain the tube holes is fusion welded with longitudinal seams to the sector having the lesser thickness. In the case of the first two of the new Conners Creek boilers, for example, the upper center drums are fabricated of $2^{15}/_{32}$- and 4-in. plate and the mud drums of 2- and $3^1/_2$-in. plate, both longitudinally welded. Despite the fact that the bottom sector of the 54-in. drum was increased to 4 in., it was necessary to swedge down the boiler-tube ends entering that sector from $3^1/_4$ to $2^{15}/_{16}$ in. OD in order to maintain ligament strength. The upper side drums have $2^5/_{32}$-in. uniform thickness shells with one as against two longitudinally-welded seams in the upper side drums and mud drums. The welds in these latter drums are located at the points where the wall thickness increases as shown in the cross-sectional elevation Fig. 19.

The drums of Boilers 1 and 2 were designed with a safety factor of five for a steel of 55,000 lb per sq in. tensile strength. With the acceptance by the Boiler Code Committee of a higher-carbon steel of 70,000 lb per sq in. tensile strength, reduction in the thickness of the drums for Boilers 3 and 4 was made possible because of the 3000 lb per sq in. increase in allowable bursting stress. Table VIII shows a comparison of plate thickness for Boilers 1 and 2 with that of Boilers 3 and 4.

TABLE VIII
SHOWING THE EFFECT OF USING HIGH TENSILE STEEL ON BOILER-DRUM THICKNESS

		Boiler Plate Thickness, 55,000 Lb per Sq In. Steel		In.—Safety Factor = 5 70,000 Lb per Sq In. Steel	
Drum Location	Drum Diam., In.	Top Sector	Bottom Sector	Top Sector	Bottom Sector
Upper center	54	$2^{15}/_{16}$	4	$2^1/_8$	$3^3/_8$
Upper side	48	$2^5/_{32}$		$1^3/_4$	
Mud	42	2	$3^1/_2$	$1^1/_2$	$2^{15}/_{16}$

An attendant advantage of the higher-carbon steel is the reduced possibility of tube-hole distortion during tube rolling. Since the swedging down of the ends of the tubes entering the 54-in. wall had not noticeably affected the water circulation, advantage was again taken of this feature in the second pair of boilers although perhaps not altogether necessary with the higher-strength steel.

The two upper side drums are each equipped with a single 4-in. Crosby high-lift pop safety valve having a discharge capacity of 238,000 lb per hr. The superheater-outlet header has one safety valve of 78,000 lb per hr discharge capacity. The use of one high-capacity valve permits a substantial reduction in the number of boiler-drum connections required by previous practice as exemplified by the Delray boiler of only slightly less steam-generating capacity where four saturated-drum nozzles equipped with Y bases were provided to accommodate eight safety valves of only about 50,000 lb per hr capacity each. Water level is maintained in the two upper side drums by Copes regulators having tension-type thermostats which respond to water-level changes without being influenced by steam flow. The regulator on the radiant-superheater side can pass up to 255,000 lb per hr and the other on the radiant-convection superheater side up to 180,000 lb per hr, a provision made necessary by the difference in superheater surface. The

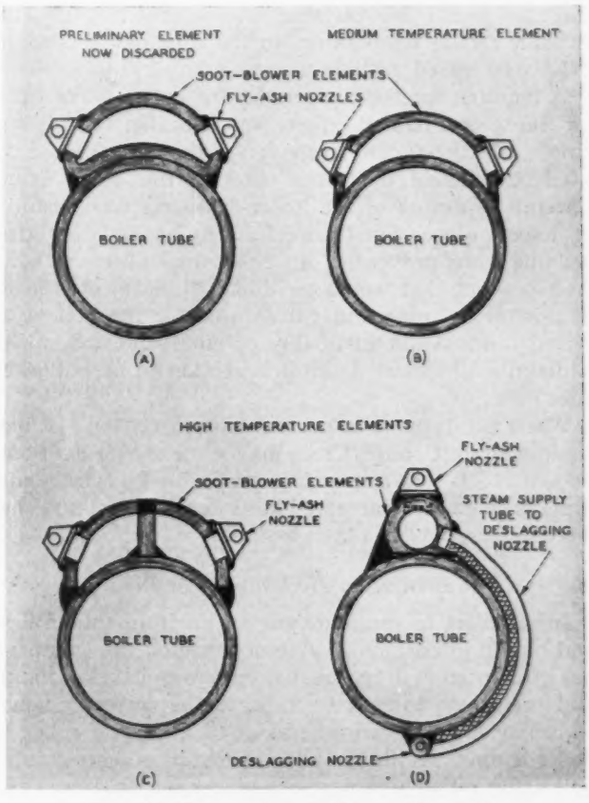


Fig. 23—Integral soot-blower elements

pressure drop across each regulator is maintained at 25 lb per sq in. through the automatic control of a pressure regulating valve to produce the same characteristic Copes-flow versus water-level curve despite a considerable variation in water pressure occasioned by different lengths of boiler-feed piping and boiler loading.

Since modern boiler design requires that shop practice be most thorough and exact, some particulars regarding inspection of the new Conners Creek boiler equipment may be of interest.

Inspection

The increased rigidity of inspection of the new steam-generating equipment is characteristic of the careful study and thought given by the Company's engineers to the whole subject of higher steam pressures and temperatures. The attitude of their inspectors reflected at all times the desire to cooperate fully with the manufacturers in obtaining a superior product with the least amount of lost time, effort and material. Unlike the policy of industry in general, inspection was not confined to scrutiny and physical test of only the finished product, but included an equally careful examination in which the manufacturer, Combustion Engineering Company, assisted after each and every major fabrication operation. As a result, many wasteful operations such as machining a faulty part were prevented, imperfections of design or processing resulting from the use of alloys never before attempted were observed at the source and promptly corrected, and a satisfactory conformity with specifications absolutely assured. Such rigid inspection, although unusual, proved fair, profitable and acceptable to all parties concerned.

Between 30 and 50 per cent of the tubes used in the boilers were carefully inspected for curvature, layout, hardness and other physical characteristics. Because variable rolling temperature in the tube mill was found to have increased the hardness of a number of tubes above that required for good rolling-in, the entire lot of tubes for the second pair of boilers was annealed before shipping. Additional care was given to the annealing of the swedged ends of the tubes entering the 54-in. drum. Careful inspection of the boiler-drum material resulted in detecting occasional imperfections immediately after bending, thus preventing an uneconomical use of labor on a product that would have had to be rejected in its final stage. Piping and valves used for the boilers received unusual but justifiably careful inspection, which is described in some detail in a section to be published later on "Piping."

While the type of inspection carried out on the new equipment for Conners Creek may savor of over-cautiousness, it is felt that a wealth of profitable knowledge and experience has been acquired in its development and that a reliable product has been assured.

The Elongation Method of Tube Rolling

In an effort to eliminate guesswork from tube rolling and obtain uniformly satisfactory results, the Company has attempted with reasonable success to develop simple and rugged equipment to indicate the correct amount of rolling for the various phases of attaching tubes to boiler drums. With the added resistance against tube-hole deformation obtained with higher carbon drum steel, the precision drilling of the holes under careful inspection, and finally with the newly developed check on tube expanding, all reasonable known precautions have been taken to insure adequate ligament strength between the tube holes. Since the elongation method for rolling-in boiler tubes has been fully discussed,[3] only a brief description need be given here.

The only extra tool required for the elongation method of rolling-in tubes is a dial indicator fitted with a proper sized magnetic or three-point-bearing clamp readily attachable to the tubes. The end of the dial-indicator shaft rests on the tube sheet or drum. A "parallel" expanding tool is used. During the first operation, when the tube is loose in the hole, the dial needle vibrates until the tube has been stretched into circumferential contact with the tube hole. During the expanding operation, the wall of the tube is squeezed between the expanding tool and the surface of the tube hole, thus forcing metal to flow axially. This axial flow causes movement of the dial-indicator needle, the amount of the movement necessary to produce the best joint being established by test. As a result of the tests so far carried out, an elongation of 0.020 to 0.025 in. is recommended for the most satisfactory joint, apparently independent of the tube and drum dimensions. Additional tests are being conducted to determine the effect of hardness and other factors which Messrs. Fisher and Cope intend to discuss in the near future.

The Compensating Superheater

The development of higher temperature power-plant heat cycles is governed largely by the availability at a reasonable price of alloys to withstand high temperature. Only comparatively recently such alloys became economically feasible for temperatures up to 850 or 900 F. Since reliability and prevention of excessive creep require, however, that alloys not be subjected to temperatures very much higher than that for which they are designed, the compensating superheater has been developed to control steam temperature as measured either at the superheater, throttle or in the main piping.

The compensating superheater installed in the new Conners Creek boilers is the outcome of experimental studies made at the Trenton Channel Plant over a number of years. The developmental history of this type of superheater was fully covered by E. V. Rieder in COMBUSTION, November 1934. Satisfactory temperature control with the finally adopted design for the experimental installation in this pulverized-fuel plant did much to encourage the use of higher superheated-steam temperatures at Conners Creek.

The compensating type of superheater should not be confused with the probably better known, though possibly less economical, by-pass type which provides an independent path around the main superheater, which path is void of superheater surface and through which an increasing proportion of combustion gas passes as the load approaches a maximum. The compensating superheater avoids the disadvantages of the by-pass type while retaining all its advantages, by providing two independent gas paths each controlled by a damper and containing, in the Conners Creek boilers for example, roughly 15 per cent of the superheater surface in one passage and 85 per cent in the other. Since, in the com-

[3] See A.S.M.E. *Transactions*, Vol FSP-57-7, May 1935, by F. F. Fisher and E. T. Cope. Also COMBUSTION, May 1935.

pensating type, the flow of combustion gases is equalized at approximately maximum rating, very little, if any, increase in fan size or in boiler and superheater surface is required.

The compensating principle is readily applicable to any type of superheater where steam temperature is elevated by the flow of hot gases across heat-transfer surface. For many years past attempts, some successful, have been made to combine radiant and convection surface so that the characteristic temperature curve of the combination would be approximately horizontal. By combining semi-radiant with convection surface, a design of variable-gas-distribution compensating superheater with a reasonably good characteristic curve was obtained, see Fig. 22. Whether or not the semi-radiant surface need be brought still closer to the fire remains to be determined by additional experimental work.

Some advantages of the compensating-type superheater are as follows:

1. The compensating superheater is peculiarly adapted to the "*W*" and "*VV*" types of boiler pioneered by The Detroit Edison Company and developed as a definite policy since 1912.

2. Instead of maintaining constant superheater-outlet temperature, the dampers can be operated to give a constant turbine-throttle temperature regardless of the variation with boiler loading or of temperature loss by pipe radiation.

3. The addition of a remote-control damper-regulating device does not complicate operation to any noticeable extent.

Integral Soot Blower

More effective elimination and prevention of slag formation and fly-ash accumulation improves the uniformity of control of the superheated-steam temperature. The development of boilers burning large quantities of coal at high rates of heat liberation has increased the tendency for slag formation on the front boiler tubes above the combustion chamber, and for fly ash to carry over within the tube banks. The greater depth of tube banks and the higher gas temperatures associated with modern boilers suggests that the whole problem of cleaning the outside tube surface, while the boiler is under load, should be attacked from an entirely new viewpoint if high steaming rates are to be maintained for indefinite periods.

Sensing that this work would have to be undertaken sooner or later, the Company's engineers began experimenting with integral soot blowers at the old Conners Creek and Marysville power houses in 1931. The blower element, consisting of a boiler tube squashed into a half-moon cross-section, see Fig. 23(A), with the inner rim of the crescent taking approximately the curvature of a $3^1/_4$-in. OD tube, was welded to the backs of certain boiler tubes located in strategic positions. Nozzles were fitted into the elements pointing in the desired directions approximately parallel with the gas flow. While the boiler was in operation the blower assembly was supplied at usual intervals with 125-lb saturated steam as required to dislodge ash accumulations, each blower element being connected at one end to a steam manifold in a position withdrawn from the hottest gas passage. At the gas temperatures found in these experimental installations, sufficient heat was transferred through the attachment

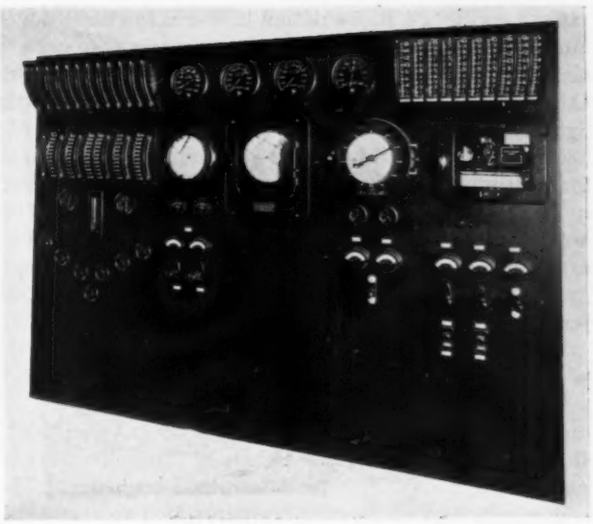


Fig. 24—Operating board of new Conners Creek boiler

to the tube wall to prevent burning of the integral elements with normal circulation of the boiler water. The results of these experiments were so satisfactory that it was decided to try out similar elements at Delray Power House No. 3 under more severe service conditions.

The temporary Delray installation made the following year on a boiler operating at 400 lb, 700 F proved that two conditions would have to be satisfied before the integral soot blower could be used with the high gas temperatures for which it was finally intended. First sufficient area of metal must be provided for conducting the heat absorbed by the element, and second, the welding between the element and the boiler tube must be ample for transmitting this heat to the tube wall and thence to the boiler contents. To satisfy these conditions, the soot-blower element ultimately took the form of two quarter sections of a $2^5/_8$-in. OD tube joined with a welded center rib, see Fig. 23(C), the whole carefully welded to the boiler tube. This rib type which was installed in No. 2 boiler in the rebuilt Conners Creek plant has proved quite satisfactory.

One successful element being found, attention was devoted to reducing fabrication costs by general all-round simplification of design. The new element, a $1^1/_4$-in. OD tube with $^1/_4$-in. wall milled flat on the side adjacent to the boiler tube, was attached to the tube with a three-bead weld to insure having a sufficient heat path. Nozzles pointing in various pre-selected directions are welded into the tube as desired. This scheme also adapted itself well to the use of nozzles on the front of the first row of boiler tubes facing the combustion chamber, see Fig. 23(D). Tube-front nozzles, which can be seen in Fig. 21, point up close to and along the tube so as to dislodge any slag forming on the combustion chamber side which cannot be reached by the regular tube-back nozzles. Deslagging of this first row of tubes in the front bank, through preventing excessive furnace temperatures under conditions of high efficiency and low excess air, has assisted materially in maintaining high boiler loads continuously over longer periods.

For superheater cleaning where the lower element is protected by a baffle, and farther back in the boiler where gas temperature is lower, a simple half-tube without any

center rib (Fig. 23(B)), a design tested in the early developmental stages, has proved a simple, economical element. In the two rear passes of the first four boilers, stationary Diamond soot blowers were installed. As the high-velocity steam simultaneously ejected from all the nozzles of the integral blowers in the front of the boiler retains sufficient energy to clean the back passes also, the stationary Diamond blowers already installed were plugged up and no further installations now seem necessary in subsequent boilers. The integral soot blower, during the limited time it has been operated, has proved sufficiently satisfactory to justify duplicating in future boilers.

Soot blowing is controlled by quick-opening globe valves located directly above the two saturated-steam boiler drums. Each valve, through a header system inside the boiler setting, simultaneously admits steam into all integral elements in one-eighth of the boiler proper. Sections are blown in a predetermined sequence. The economizers and air heaters are controlled similarly but by a separate system.

The economizers and plate-type air heaters are equipped with one element each in the gas inlet. By diagramming, egg-crate fashion, the area of the gas entrance passage and locating one nozzle in each small area, one stationary grid element is designed to blow each appliance. It is believed that by this arrangement the steam jets mutually restrict the spreading of the jet of steam beyond the boundaries of each "egg-crate," thus maintaining the steam velocity so as to do effective cleaning at the far end of the equipment.

Saturated steam at about 650 lb per sq in. boiler pressure is supplied directly through the control valves into the soot-blower headers. A marked improvement in the appearance of the boiler is accomplished by placing all soot-blower piping within the setting, thus eliminating conspicuous condensate traps and attendant unsightly piping. Any condensate on the pressure side of the control valve drains by gravity back to the steam drum. The soot-blower piping and elements are designed to take up about 550 lb per sq in. in friction loss so that the steam pressure at the nozzles is approximately 100 lb per sq in. There is a minimum change in boiler-water level as each section of the steam-generating equipment is blown for only ten seconds.

Soot disposal is a further interesting refinement of design incorporated in the new Conners Creek boilers. By a series of internal ducts and power-driven feeder valves which can be distinguished in Fig. 19, all the soot hoppers, including those under the economizers and air heaters, and the rear passes of the boiler can drain into the stoker hopper from which the soot is fed into the furnace for burning. Incorporated with these drains are four-way valves to direct when necessary the discharge of cinders from either the front or the rear mud-drum hoppers to either the stoker hopper or to a water-drain connection to the basement for use when washing down economizers. As the four-way valves prevent short-circuiting the draft in the boiler passes, either the front or the rear mud-drum hoppers, as desired, may be continuously discharged to the stoker. Operation so far has been gratifyingly satisfactory.

Forced and Induced Draft Increased

The increased steam-raising capacity of each boiler space, as already described, required sources of draft of both greater capacity and higher static head than could be produced by the old equipment. Naturally induced draft up to about $1^1/_2$ in. was sufficient for the old boilers since economizers were never installed although space had been provided. The original forced-draft fans, installed in the pipe gallery below the boiler-room floor, supplied up to 74,000 cfm of air at 7-in. water pressure to the plenum chamber of each boiler. These latter were replaced with forced-draft fans of 101,000 cfm capacity (at 29.92 in. Hg, 68 F and 50 per cent relative humidity) with a static head of up to 15.78 in. of water to supply the draft loss occasioned by the air heaters, zoned combustion-air control and the considerably higher rate of combustion.

Because of the greater friction loss associated with larger gas volume passing from the new boilers through the old stacks, and the addition of economizers and air heaters, coupled with a much lower stack temperature available for creating natural draft, induced-draft fans were installed with the new steam-generating plant. These fans, like the stokers, are driven by 230-volt dc motors to obtain the desired flexibility of operation. The new induced-draft fan is designed for 181,000 cfm capacity (at 29.92-in. Hg and 346 F) at a static head of up to 15 in. of water. Both the forced-draft and induced-draft fans used with the new boilers were made by the American Blower Co.

Little alteration was necessary in the stack proper. The uptake from the old boiler outlet to the base of the stack was removed. The new breaching from the induced-draft fan, located on a new floor $16^1/_2$ ft above the old belt-conveyor floor, is specially designed to regain some part of the velocity head of the gas leaving the fan outlet. A wide sweep is provided to deflect the gas into the stack with a minimum head loss. The available natural draft is reduced both by the new stack gas temperature, lowered by about 400 F, and by a greater friction loss in the stack as the result of higher gas velocity. Except for regular maintenance, nothing more had to be done to the stacks to fit in the new boilers.

Gage Board

Of the remaining boiler equipment not already discussed, brief mention should be made of the boiler-operating gage board if only to point out the comparatively simple instrument layout required for the air control. As will be seen from Fig. 24, the new gage board is marked for its neat appearance and arrangement. The left-hand third of the board is devoted to air-flow and damper-position indicators, air- and water-pressure gages, gage-manifold switches and zone-loading controls. To an operator familiar with the board, this equipment tells at a glance the exact condition of the fire on both stokers of the boiler.

Of the other instruments in the remainder of the board, little need be said. Because the new steam plant has more equipment, there are more draft gages. Steam-flow air-flow meters are on both the old and new boiler panels. The new installation includes far more elaborate and exact temperature-indicating and -recording instruments for maintaining a nearly constant superheated-steam temperature, and the keeping of full operating records. The use of remote motor control has removed the unsightly drum controllers. It is felt that the present layout reflects the latest experience and is very satisfac-

COAL HANDLING AT CONNERS CREEK

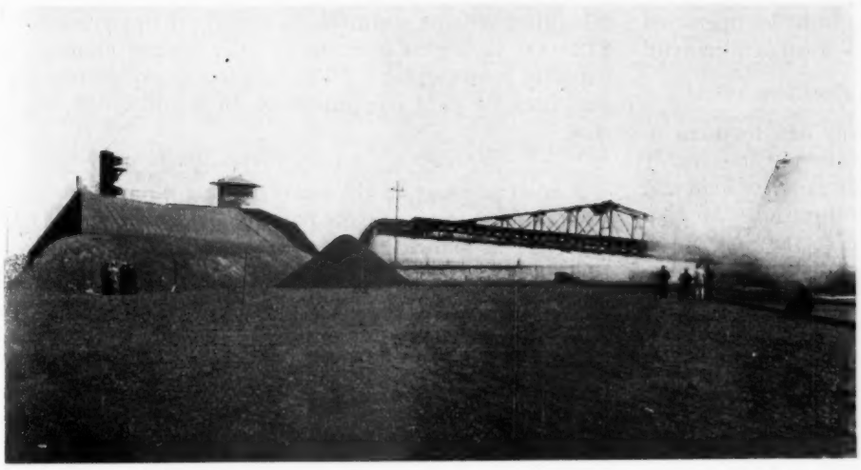


Fig. 25—Unloading first cargo of water-borne coal at Conners Creek dock

Fig. 26—Reclaiming coal from pile on dock

Fig. 27—General view of yard layout at Conners Creek

tory for the somewhat new variations in boiler operation created by the newer design of steam-raising equipment.

Boiler Performance to Date

The particular objective of the many new features incorporated in these boilers was to make possible highly efficient operation through a very wide range of load and uninterrupted runs of several months' duration. Air control and superheat adjustment relate to the first objective; special cleaning features are intended to accomplish the second.

Operation over a period of several months indicates the first objective has been accomplished to a very satisfactory degree. Variations in coal and ash characteristics, and the high furnace temperature resulting from high CO_2 operation have made the second objective difficult to accomplish. The special cleaning features have tended, however, to shorten the cleaning time and reduce outage materially.

Coal Handling and Preparation, and Ash Removal

Fortunately the existing coal-handling and preparation equipment is still in excellent condition and capable of operating a few more hours per day to supply fuel for a plant of double the original size. According to usual practice the original equipment was selected with a view to providing sufficient capacity for preparing and delivering to the bunkers during a short shift, enough crushed coal to carry the plant through the 24-hr period. This capacity in conjunction with ample bunker storage makes it possible to prepare and deliver to the bunkers, still during a single shift, a sufficient supply of crushed coal to carry the new boilers through a period somewhat in excess of 24 hours even when operating at maximum rating.

Continued Use of Existing Coal Equipment

The availability of this equipment for continued use, to all intents and purposes just as it stood, had in two ways a considerable bearing on the rebuilding plans. First, it was possible to double the capacity of the power plant without making a corresponding outlay on equipment for preparing, handling and storing crushed coal. This fact, coupled with similar ability to re-use the building, canals and foundation work for double capacity with only slight additions, tends toward continually decreased investment costs per installed kilowatt as the rebuilding proceeds. The rebuilding plan avoids in this respect a considerable duplication in investment which would occur if the additional capacity were placed in an entirely new plant located on an adjacent site. Second, the fact that the equipment was suited for stoker firing rather than for using pulverized coal did much to swing the choice in favor of the former method.

The value of the coal equipment suitable for stoker use only, plus the portions of the building structure required to support it and to confine dust to the conveyor space, or in other words, those items not suitable for reuse with a pulverized coal system, amounted to about $350,000. The coal-unloading house, receiving hoppers, coal crushers and apron conveyors from hoppers to crushers represent another $375,000 investment which could be utilized in rebuilding the old plant for either pulverized coal or stoker firing, but which would have to be duplicated if an additional plant were built on an adjacent site. Thus, in the matter of coal supply alone, the rebuilding scheme is entitled to a credit of approximately $725,000 saving in investment over adding equivalent capacity in a new plant. These figures do not involve the coal dock or yard equipment which would do in either case.

Capacity of Coal-Handling Equipment

A brief account of the coal-handling equipment may be of interest in tracing the path of coal from railroad car to stoker as shown in the old and new plant cross-sections, Figs. 3*a* and 3*b*, respectively.[4] Coal, after being weighed on railway scales, enters the unloading house in hopper, or in drop-bottom cars. Parallel tracks can be used if necessary to unload two cars simultaneously into the same hopper. There are four hoppers under the tracks in the unloading house, from which the coal is elevated by four Link-Belt Co. "apron" conveyors to four Williams crushers where it is reduced to standard size. Crushed coal is hoisted in four Peck bucket conveyors to the top of the boiler house where the buckets are tripped on to three Robins belt conveyors extending the length

TABLE IX
COAL DEMAND FOR REBUILT PLANT

	330,000-Kw Plant, 12 Boilers		390,000-Kw Plant, 14 Boilers	
	Good Coal	Poor Coal	Good Coal	Poor Coal
Coal burning rate, lb per kw hr*	1.0	1.14	1.0	1.14
Hourly coal demand at rated plant capacity, tons	165	188	195	225
Hourly capacity of coal equipment, tons**	480	480	480	480
Daily coal demand at 100% plant use factor, tons	3960	4512	4680	5400
Daily coal demand at 75% plant use factor, tons	2970	3384	3510	4050
Daily coal demand at 50% plant use factor, tons	1980	2256	2340	2700
Hours' operation coal equipment per day, 100% plant use factor	8.25	9.40	9.75	11.25
Hours' operation coal equipment per day, 75% plant use factor	6.20	7.05	7.31	8.45
Hours' operation coal equipment per day, 50% plant use factor	4.12	4.70	4.87	5.62

	Any Size Plant	
	Good Coal	Poor Coal
Available bunker storage per boiler, tons	500	500
Time required to exhaust available bunker storage, hrs		
30,000 kw average load per boiler	33.3	29.2
25,000 kw average load per boiler	40.0	35.1
20,000 kw average load per boiler	50.0	43.9
15,000 kw average load per boiler	66.7	58.4
10,000 kw average load per boiler	100.0	87.6

* Assuming an average plant steam rate of approximately 10 lb per kw hr, purposely chosen somewhat high.
** This is rated capacity. Actual test shows up to 543 tons per hour.

[4] See Combustion, November 1935.

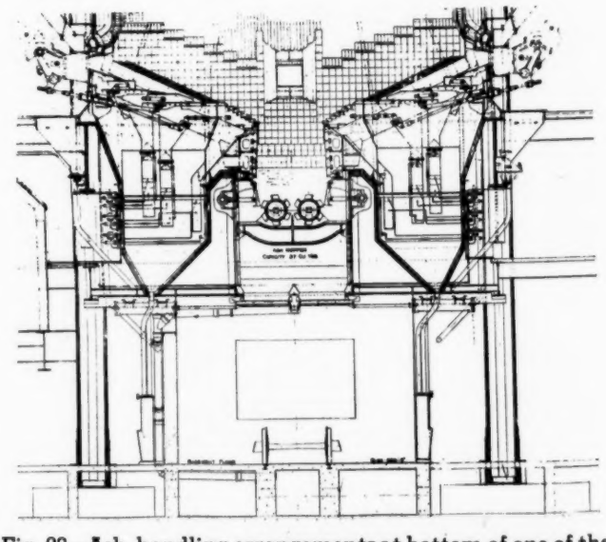


Fig. 28—Ash-handling arrangements at bottom of one of the new boilers

of the three rows of bunkers, from which it feeds as required through spouts and spreaders to the stoker hoppers. Each of the four sets of equipment enumerated above has a rated capacity of 120 tons per hour, giving a total capacity for the plant of 480 tons per hour. Actual tests of the equipment, however, have shown a capacity when operating continuously of 543 tons per hour which is fully equal to the demands of the double-size rebuilt plant as shown by the accompanying tabulation.

The foregoing data show that while the bunker capacity is ample for day-to-day storage, it is somewhat short of providing sufficient margin for week-end carryover, especially when there is a contiguous holiday. The present tendency toward base-loading the few new boilers and turbines initially installed accentuates this shortcoming and makes it necessary to refill the bunkers once or twice during a week-end. This, however, is a minor inconvenience and it is doubtful whether larger bunkers would be justified if the plant were being built new.

Rail and Water-Borne Coal

Until 1930 coal was received at Conners Creek by rail only. Entering cars were either unloaded into stock by locomotive cranes having grab buckets, or dumped through their drop bottoms into hoppers underneath the tracks in the coal-unloading house. Over two track hoppers is located an unloader consisting of an overhead traveling crane equipped with heavy spuds for breaking up masses of frozen coal in the cars. The unloader also serves to move cars along the track inside the unloading house. Coal from storage is reclaimed by locomotive cranes loading into railroad cars, which are then pushed by a yard locomotive to the unloading house. It has been the Company's policy to keep a one to three months' supply of coal on hand at all times to guard against a temporary stoppage in delivery.

In 1930 equipment was installed for handling water-borne coal to take advantage of the favorable water freight rate from the lower lake ports. Self-unloading vessels dock on the river frontage and discharge their cargoes on to the shore, Fig. 25, within reach of a Beaumont drag scraper where about 20,000 tons can be piled at one time. The arrangement for reclaiming coal from this pile was described in an article by J. R. James in *Power* for November 3, 1931. Cars loaded by the drag scraper, Fig. 26, are removed by a yard locomotive. Hopper-bottom cars are used for coal sent to the coal-unloading shed. Flat-bottom cars are sent to the stock piles for unloading by locomotive crane.

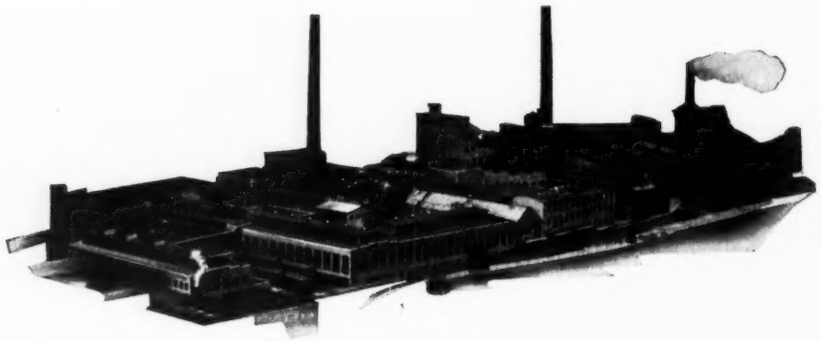
Navigation on the Great Lakes is open for approximately eight months of the year, during which period all coal received is water borne. During the autumn the stock piles are built up sufficiently to carry the plant through the winter. For several years no coal has been received by rail, but, as the plant capacity approaches 300,000 kw, it may become necessary either to revert in part to rail-borne coal during the winter months or else to increase the acreage of the storage piles. On the 67-acre tract on which the plant is situated, there remains sufficient space for stocking a large supply of coal. With the yard trackage arranged as at present, it is possible to stock up to 175,000 tons, counting the 20,000 tons on the coal dock. By rearranging the tracks and only slightly increasing the space, it would be possible to augment the storage to 270,000 tons, still using the locomotive-crane method of handling into and out of stock. A general idea of the yard layout can be obtained from the photograph of Fig. 27.

If more coal storage is desired it may eventually be necessary to clear that part of the property now used for cable drums, unsold scrap equipment and so on. Experimentation during recent years indicated that the most economical height from the point of view of loss by spontaneous combustion is thirty feet from ground to apex of the coal pile, which result was definitely incorporated in the rules for stocking coal. Some idea of the amount of storage which will be required can be gathered from the daily requirement of a 330,000-kw plant operating at 50 per cent use factor being in the neighborhood of 2000 tons. Navigation in the Detroit River is reckoned to be closed from December 1 to March 15. At the end of the closed season, policy dictates that there shall remain further storage sufficient to supply the plant from one to three additional months depending upon the indications of labor unrest and future business. If a 136-days' supply is to be on hand at the close of navigation in the Fall for carrying through the closed season, the storage of about 270,000 tons required for a 330,000-kw plant can be provided for, as explained, quite easily. Allowing for an additional 61 days, or 197 days in all, about 400,000 tons must be on hand. This can be suitably provided in the cleared property, still using the same methods of storage. The advisability of increasing the storage to this extent will depend on the differential between rail and water freight rates at the time.

Ash Handling

Ash is removed from the boilers in standard railroad cars entering the boiler-house on tracks at grade level. As shown in Figs. 3*a* and 3*b* (see the first article), and in more detail in Fig. 28, an ash aisle is situated along the center lines of the two rows of boilers below the mezzanine floor. The existence of these aisles with their standard track, and the possibility of incorporating an adequately-sized ash hopper for the new boilers and still leave head room for the standard steel cars were also favorable to a rebuilding program using double-ended stokers rather than pulverized coal. A further consideration in favor of stoker firing is the profitable market for the sale of cinders which has existed since the plant was started.

Ash discharged by the clinker grinders is held in large hoppers and periodically dumped directly into the cars. In order to accommodate the new 37-cu yd ash hopper in proper relation to the new extension grates and ashpit, it was only necessary to reduce the clearance above the tracks to 11 ft, leaving about $1^1/_2$ ft less than in the old plant. The extension grates and the ash hopper are provided with general service water sprays for quenching and washing. The ash-dump gates are of the air-operated sliding type and are lined on the upper side with gunite. The ashes are dumped once or twice every eight-hour shift, and upon opening of the gates fall by gravity into the cars placed below. The cars are drawn out of the aisles by coupling with other cars attached to a yard locomotive, and are emptied by the grab-bucket crane which also piles the cinders on the property adjacent to a trucking driveway. Ash is filled into contractors' motor trucks by the same method and hauled away for grading parking lots and cinder drives, and for building construction.



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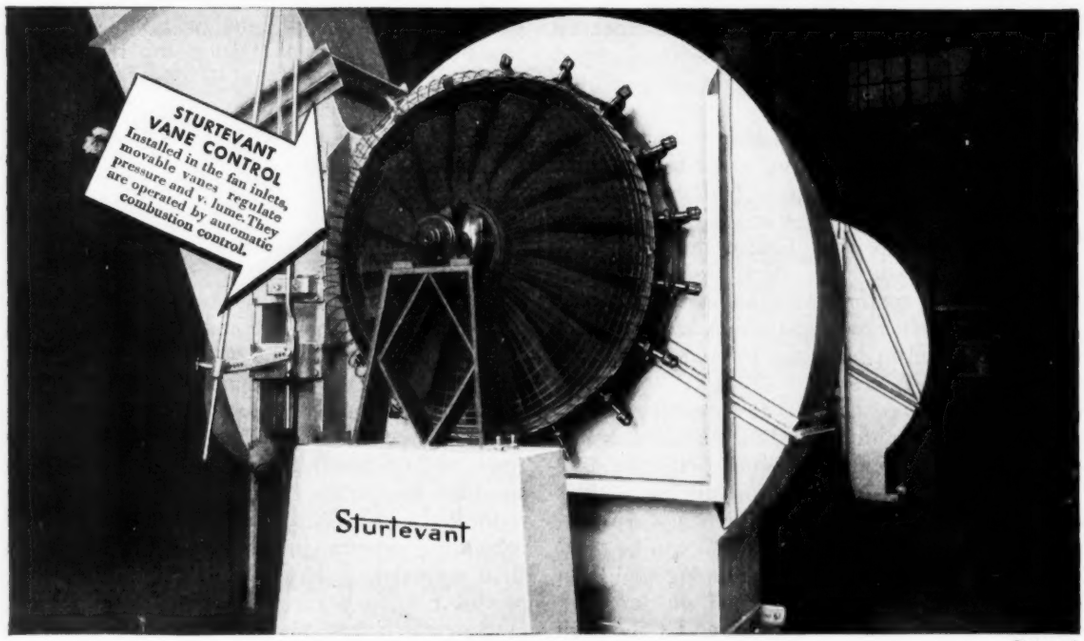
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JAMES WATT, Engineer and Scientist

(January 19, 1736 - - August 19, 1819)

A brief review of Watt's contribution to the machine age through his transformation of Newcomen's crude and wasteful pumping engine into an economical source of power. While not the inventor of the Cornish boiler, he did much to improve its primitive form and, by patient experimentation, to establish certain fundamental data relative to the generation of steam. An insight into his personal characteristics and business acumen is shown in his relations with such men as Boulton and Roebuck.

By H. R. TAUBE*

MANKIND is celebrating this month the 200th anniversary of the birth of one of its benefactors, James Watt, who was born in Greenock, Scotland on January 19, 1736. It is the popular belief that Watt was the inventor of the steam engine, and that his invention was the starting point of the Industrial Revolution, that period of quick economic and social changes which brought about modern industrialism. Like most attempts to compress into a brief formula the meaning and content of a complicated sequence of events, this belief is not quite in keeping with the facts.

Watt's statue in Westminster Abbey, erected by public subscription in 1824, bears on its pedestal an inscription composed by the great Lord Chancellor, Lord Brougham. It praises Watt as the man who directed "the force of an original genius, early exercised in philosophical research, to the improvement of the steam engine." As a matter of fact, when Watt, in 1769, received his first patent for "lessening the consumption of steam and fuel in fire engines," these engines, invented by Thomas Newcomen, a blacksmith, had been in common use for two generations. The fire engine was based on a principle discovered by the French protestant refugee, Dr. Denis Papin, who had described it in a paper published in 1690. The engine consisted of a cylinder open at the top and of a piston sliding along its inner surface. The piston was connected by a chain with the end of a beam swinging about its middle and bearing at its other end a counterweight, able to raise the piston. Steam introduced under the piston expelled the air and, when cooled by a jet of water, condensed, creating a partial vacuum which permitted the pressure of the atmosphere to force the piston downward.

Watt himself has described the defects of Newcomen's engine and modestly appraised his own basic invention as follows:

"When analyzed, the invention would not appear so great as it seemed to be. In the state in which I found the steam engine it was no great effort of mind to observe that the quantity of fuel necessary to make it work would forever prevent its extensive utility. The next step in my progress was equally easy—to inquire what was the cause of the great consumption of fuel. This too was readily suggested, viz., the waste of fuel which was necessary to bring the whole cylinder, piston and adjacent parts from the coldness of water to the heat of steam no less than from 15 to 20 times in a minute."

During a walk, one Sunday afternoon, in June 1765, it became clear to him that

"as steam was an elastic body, it would rush into a vacuum, and, if communication were made between the cylinder and an exhaust vessel, it would rush into it and might be there condensed without cooling the cylinder. I then saw that I must get rid of the condensed steam and injection water if I used a jet as in Newcomen's engine."

The "exhaust vessel" mentioned by Watt is the condenser, the use of which eliminated much of the thermal losses in the cylinder. Further, by enclosing the cylinder in a larger vessel filled with live steam, the latter replaced the atmospheric pressure for forcing the piston down and cooling of the cylinder walls by the air was avoided. The result was that an engine built according to Watt's patent would save three-fourths of the fuel required by a Newcomen fire engine of the same capacity, but otherwise the working principle, appearance and range of usefulness were the same as that of its predecessor. It could pump water, blow the bellows of a blast-furnace, but it could not turn the wheels of industry. To do that a rotative engine was required which Watt developed and which was put on the market in 1783.

At the time of its appearance many branches of British industry, in the first place cotton spinning, but also milling, brewing, the metal industry and mining, had long ago outgrown the handicraft stage and represented capitalistic enterprises carried on in large establishments employing large numbers of workmen. The only source of mechanical power then available to industry was, besides the muscle power of men and animals, the energy of falling water, and this required in many cases the erection of factories and works in inaccessible, out-of-the-way places.

As roads were scarce and bad, the inconveniences and expenses caused by the transportation of materials and manufactured goods, and by the housing and feeding of numerous employees, were considerable. The steam engine broke the tie, chaining industry to sites of water power, providing an economical and illimitable source of motive power.

James Watt†

* Member of the A.S.M.E.

† Reproduced from Thurston's "History and Growth of the Steam Engine," published by D. Appleton-Century Company.

Watt's rotative, double-acting expansion engine which was patented in 1782, answered a crying need, because industry had nearly exhausted the available sources of water power, and its further growth and development would have been stunted and retarded by lack of motive power. On the other hand, the immediate success of the steam engine was due in no small degree to the progress of mechanical technology made shortly before its advent, which had made available means to produce the large, comparatively accurately machined castings Watt's engine required. A few decades earlier this would have been impossible, and Watt owed much to the great iron master John Wilkinson, the inventor of the cylinder-boring machine, whose works for many years supplied the firm of Boulton & Watt with all heavy castings.

Watt was not less fortunate in having as his financial backers and associates two of the most prominent businessmen and manufacturers of his time. The first of them was Dr. John Roebuck, the part-owner and manager of the Carron Iron Works, near Linlithgow, Scotland, who needed an efficient pumping engine to keep in operation his flooded coal mines. He paid Watt £1000, undertook to finance further experiments and the manufacture of his engines, and provided the means for obtaining a patent. On his way to London to apply for a patent, Watt visited in Birmingham Matthew Boulton, a wealthy manufacturer of buckles and brass buttons, which in the 18th century were generally worn and were in great demand, of silver-and gold-plated ware and ornamental iron and bronze work. His factory at Soho, near Birmingham, was famous the world over for its modern equipment and exemplary organization, and its owner was so highly esteemed that the Empress of Russia, Catherine the Great, while traveling in England, inspected the factory and stayed for several days at Boulton's house. Boulton was mildly interested in Watt's invention. Soho was suffering in summer from a lack of water to drive its water-wheel, and Boulton had thought of installing a pumping engine for raising the water from the mill stream and returning it back into the mill pond. He had sought the advice in this matter of Benjamin Franklin who was at that time in England on a diplomatic mission for the Assembly of Pennsylvania, and had been in correspondence with that great American.

The modest young inventor from the backwoods was deeply impressed by the grand manner of the magnificent merchant prince who, in a general way, expressed his willingness to acquire a share in the patent. On his return to Glasgow Watt wrote Roebuck about his interview with Boulton, trying to impress upon him the advantages of this prominent man's collaboration. At the same time one more experimental engine, with a cylinder of 18-in. diameter and 60-in. stroke, cast of block-tin, was built at the Carron Iron Works but it was unsuccessful. Despite Roebuck's urgings, Watt did practically nothing to make the necessary changes and adjustments, and for several years neglected his invention completely.

He knew evidently that Dr. Roebuck had suffered severe losses and that he was heavily in debt. He might have feared that in case of bankruptcy a successful invention might fall in the hands of the creditors and he might be the loser. Meanwhile Watt remained in touch with Boulton, trying to keep awake his interest in the steam engine, and when Roebuck, in 1772, became insolvent, he received from Boulton power of attorney to obtain for him Roebuck's share in the invention. The creditors, believing the patent perfectly worthless, were only too eager to accept Boulton's offer to waive a claim for £1200 he had against Roebuck in exchange for this share.

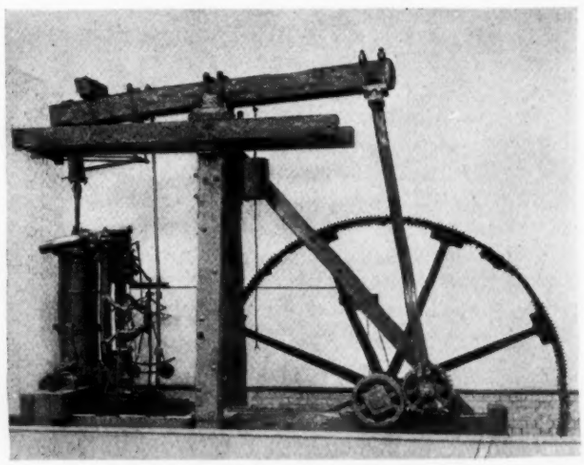
Watt's long cherished ambition to become associated with Boulton was now fullfilled. In 1775 he moved to Birmingham to become a partner in the newly established firm of Boulton & Watt, Engineers, in which Watt had an one-third interest and had charge of design, development and installation of the engines. Boulton furnished not only the considerable financial means to start the manufacture of the steam engine, but also his exceptional organizing and commercial ability, without which the undertaking would probably have failed. Before the business became profitable he had sunk into it the sum of £40,000.

Boulton had from the very beginning of his connection with Watt grasped the enormous possibilities of his invention as a source of industrial power and he urged, therefore, the development of a rotative engine for industrial purposes. Watt, who did not possess his partner's broad vision, insisted that the outlook for further industrial expansion held so little promise, that it did not justify the expenses and trouble involved in the development of a rotative engine.

As late as 1782, when the rotative engine was already patented, Watt wrote to Boulton who was away on a business trip:

"If you come home by way of Manchester, please not to seek for orders for cotton mill engines, because I hear there are so many mills erected on powerful streams in the North of England, that the trade must soon be overdone and consequently our labor may be lost."

When Watt began to develop the rotative engine he found to his consternation that the simplest method of transforming reciprocating motion into a rotative one, the crank and fly-wheel, had been anticipated by John Wasbrough and that this device had been patented. It was Watt's contention that he had worked on this idea and that it had been betrayed by one of his workmen. He has never been able to substantiate this claim which is probably unfounded. To circumvent the use of the crank, he patented in 1781, five different methods for rotative motion, though only one of them, the sun- and planet-motion, suggested by William Murdock, one of Watt's foremen, was ever used.



Watt's double-acting rotative engine

By 1785 the business of Boulton & Watt had become profitable and the period of their financial worries was over. At about the same time the development of the steam engine in its main aspects was finished, and later on only minor improvements and changes in the design of some parts were necessary. When Watt retired from business in 1800, his engine contained most of the essential features of the modern reciprocating steam engine, though in general arrangement it still retained a striking family semblance with its ancestor, Newcomen's fire engine.

The boiler being a basic element of any steam power installation, one should expect that Watt had contributed to its development, and it is, at first sight, somewhat surprising to find that he did nothing of the kind.[1]

The boilers used with the first engines manufactured by Boulton & Watt were of the beehive or haystack type, which had been in common use with the fire engine, and which were an adaptation of the brewing pan. These boilers had a cylindrical or slightly conical body, a hemispherical top, a bottom dished upward, were equipped with manhole and safety-valve, and were, as a rule, made of copper.

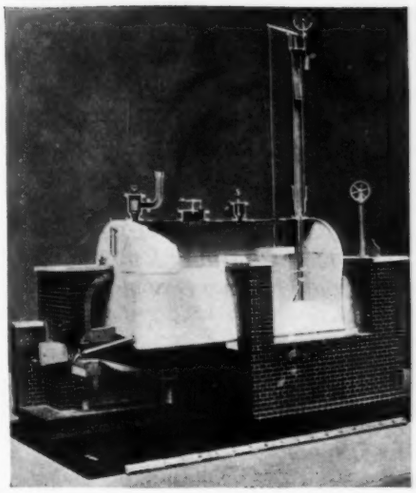
Among Boulton & Watt's first customers were the Cornish copper mines, and the Cornish engineers preferred horizontal boilers with internal flues; they were superior to the haystack type, and were adopted by Boulton & Watt for most of their later installations. Watt himself has disclaimed any credit for the invention of the Cornish boiler, which has been ascribed to him, stating that,

"the conveying of flame through flues in the inside of the water had been practiced by others before my time and was common in the Cornish engines. The inventor is unknown, but a person of the name of Swaine was a great propagator of the practice."

He added that he, Watt, had "somewhat improved the form and adjusted the proportions" of the Cornish boiler. These boilers had a flat bottom and flat sides, tied with stays, a curved top and one or two flues of round, elliptic or rectangular section. The boilers and flues were made of copper or iron. Their principal dimensions varied widely, though a boiler 19 ft long × 8 ft wide × 8 ft high, with two flues of 22-in. diameter, seems to have been used frequently.

Boilers furnished around 1790 were equipped with a dome, or "water trap." From the beginning there were difficulties in preventing boiling over and priming. Boulton & Watt published in 1779 a book of "Directions of Erecting and Working of the Newly-Invented Steam Engines," which says that the feed pipe is to be fitted with a valve "to prevent the water ever being forced up through the steam." The water trap served also the purpose of increasing the steam space, as in many cases the boilers proved too small for the engine. In calculating the necessary heating surface Watt used at different times different ratios of heating surface to evaporation. In Cornish boilers this ratio varied from 9 to $10^1/_2$ sq ft of heating surface per cubic foot of water to be evaporated per hour. The figure which for some time was adopted by him as standard was 8 sq ft of heating surface per cubic foot of water evaporated per hour.

[1] See H. W. Dickinson and Rhys Jenkins, *James Watt and the Steam Engine*.—the Memorial Volume prepared for the Committee on the Watt Centenary Commemoration at Birmingham, 1919; Oxford, 1927. The data concerning the boilers in Watt's time are based on the study by the authors of the original business records of Boulton & Watt which, after the dissolution of the firm in 1850, were carefully preserved.



Watt's wagon boiler

The furnace was built so that the gases swept the bottom sheet to the back and returned to the front through a side flue, then crossed to the opposite side and to the chimney

Early Boilers of Copper

Boulton & Watt preferred, and for good reason, copper as material for boilers. The first cost of a copper boiler was, of course, higher but the old boiler could still be sold at a good price. In 1777 a London coppersmith charged 14 pence per pound for new boilers, but gave $7^1/_2$ pence for old ones. Iron boilers cost, at the same time, £40 per ton, so that, allowing for the smaller dimensions and weight of the copper boiler of the same capacity, and for the fact that the latter would last much longer than the iron boiler, a copper boiler was the better investment.

The iron plates obtainable at the time Boulton & Watt started their business (1775), were not rolled but hammered out and, when made of English iron, were of very low quality. It might be mentioned that the British iron industry in the first half of the eighteenth century produced not more than one quarter of the total consumption of the metal, on account of the scarcity of charcoal for smelting, and that the balance was imported from Sweden and Russia. A process for using coke for smelting iron ore was invented in 1735, and half a century later the puddling process. These inventions resulted in a vigorous development of the British iron industry, but the quality of its products was so low, that Watt used to insist upon forgings made from "Russian slabs." The largest hammered iron plates were 36 × 21 × $^1/_8$ in. thick, or 26 × 21 × $^1/_4$ in. thick. In 1779 rolled plates of larger dimensions, but of very low quality, were obtainable, so that many customers continued to demand hammered plates as being of better quality. As late as 1786 Boulton wrote about some iron plates furnished by a certain iron master: "they will not bend even hot, they being made of his own bad iron."

The transportation of a finished boiler was in most cases out of the question, and the plates had to be put together in the field by local smiths. The above-mentioned book of "Directions" contains detailed instructions to this effect, one of which might be quoted: "All the joints should be wetted with a solution of sal-armoniac in water, or rather, in urine, which by rusting them will help to make them steam-tight." The rivet-holes were punched by hand on the spot, one inch from the edge of the plate and spaced two inches. The rivets were also forged in the field and it was regarded as good practice to make them twice the thickness of the plates they were to hold together.

The boilers were fed by gravity from a vertical pipe, supplied from the hot water pump. The feed pipe had to extend 7 ft above the water level in the boiler to overcome the steam pressure, and the regulation of the flow was done by a cock in the pipe. Watt had invented a glass water gage, but as it was difficult to obtain reliable glass tubes, he continued to use gage cocks. Another of his inventions was a mercury manometer of the barometer type. The "Directions" prescribe that a pressure sufficient to support a column of mercury 1 to 2 in. high should be sustained; higher pressures would tend to open the joints and to cause steam leakages, and they "do not serve any good purpose."

By 1790 boiler making had become a separate industry, and Boulton & Watt found it convenient to order their boilers outside, usually paying 36 shillings per cwt and reselling them at 42 shillings per cwt.

The primitive state of the art of boiler making and the low quality of the materials available explain sufficiently Watt's reluctance to abandon the established routine in boiler design. Any such attempt, as, for instance, the introduction of higher pressures, would have been under existing conditions a hopeless undertaking. Nearly a century of technological progress was necessary before any bold departure from the accepted practice of boiler design could be introduced with reasonable chance of success.

A more promising field was the improvement of the existing furnaces, and Watt patented in 1795 a smokeless furnace. While he was full of ideas, these ideas were not necessarily always his own. In this case he had made use of a suggestion by Benjamin Franklin to Boulton in a letter written in 1766. This letter is so remarkable for its writer's clear understanding of the problem that it is worth quoting:

"I would only repeat to you the hint I gave of fixing your grate in such a manner as to burn all your smoke. I think a great deal of fuel will then be saved, for two reasons. One, that smoke is fuel, and is wasted when it escapes uninflamed. The other, that it forms a sooty crust on the bottom of the boiler, which crust not being a good conductor of heat, and preventing flame and hot air coming into immediate contact with the vessel, lessens their effect in giving heat to water. All that is necessary is to make the smoke of fresh coals pass descending through those that are already thoroughly ignited. . ." Watt's smokeless furnace was tried out in Soho with good results, but seems to have been later neglected.

While the development of the steam engine is Watt's principal and generally recognized achievement, another hardly less important contribution to human progress has been made by him. He demonstrated the importance of science and of scientific methods in the solution of technological problems. The influence of Watt, who might be called the first Mechanical Engineer in the modern sense of the word, upon the further development of engineering and scientific research cannot be overestimated. He was endowed with the inquisitive mind, the thirst for exact knowledge and the critical sense of the true scientist, and he possessed, a rare combination, the gift of intuition, as well as a capacity for painstaking attention to the minutest details.

The way in which he solved the paradox of Newcomen's engine which, for efficient operation, required that its cylinder should be hot and cold at the same time, gives an interesting glimpse of Watt's mind in action. The idea of the condenser came to him in a flash of intuition, but this flash had been preceded and was followed by years of patient experimentation and study of the then available literature on the properties of steam. To appreciate duly Watt's feat we must remember that at that time the science of thermodynamics was not yet born and that Stahl's abstruse theory of combustion, according to which during combustion a body, called phlogiston, is liberated by the burning body, was then generally accepted. Watt had to determine experimentally such basic data as, for instance, the volume of steam generated from a given quantity of water; the quantity of water evaporated by a pound of coal; the steam pressures corresponding to different temperatures of steam.

Watt's Education and Associations

Watt's formal education had been limited to attending grammar school. After a brief apprenticeship he became, at the age of twenty-one, instrument-maker to the department of natural philosophy, as physics was then called, at the University of Glasgow. The head of the department was Dr. Joseph Black, famous as the discoverer of latent heat and carbon dioxide. The frequent contacts with this prominent scientist who took a warm interest in Watt, and with other members of the faculty, among them Dr. Adam Smith, of Wealth of Nations fame, and the access to the university library offered Watt a splendid opportunity to acquire a thorough knowledge in a wide variety of scientific subjects. This enabled him, in the later years, to move as an equal in the distinguished circle which gathered around Boulton, and men, like Sir Walter Scott, have expressed their admiration for his great intellectual versatility and gift for brilliant conversation.

Watt's human weaknesses and shortcomings cannot detract from his greatness as an inventive genius of the first order. Nor can the fact that any great invention is not so much the creation of the individual inventor, as an organic growth, the product of collective effort, lower our high estimate of Watt's great achievement: of having transformed Newcomen's wasteful pumping engine into a versatile, efficient and economical source of power. The steam engine was the offspring rather than the parent of the Industrial Revolution; but the advent of the machine age, the locomotive, the mechanically propelled ship and the development of heavy industry, the industry to make the raw materials and the machines to make machines, all this is unthinkable without steam power.

Watt and his contemporaries did not grasp the implications, nor could they have foreseen the tremendous, all-pervading sociological consequences of his inventions. Power was a new, unprecedented factor in human history and has deeply influenced its course. The steam engine has set in motion forces which man has not yet learned to control, and the world crisis of our days might, in the last analysis, be traced back to the shifts and stresses caused by its impact in the social-economic structure of Western civilization.

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Efficiency of a Reaction Turbine Stage from Static Model Tests with Air

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In model testing with air precise measurements are possible, the testing equipment can be made relatively simple, and the observers may work directly with exposed jets without personal danger. However, the fact that all conditions of mechanical similarity cannot always be satisfied makes necessary the application of knowledge based on experience. As to the two methods of determining velocity coefficients, namely, the reaction test and the impact traverse test, the author explains the limitations of the former and shows how the latter is better adapted for accurate determinations over the entire range of velocity ratio.

THE reaction turbine, or more properly, the fifty per cent reaction turbine, since the enthalpy change per stage is divided equally between the stationary and moving elements, consists of a number of rows of blades of varying height mounted on a drum or disks of varying diameter rotating within a cone containing a like number of rows of stationary blades. In this type of turbine, blades of identical cross-section are used in the stationary and moving rows and for this reason it is frequently called a turbine with symmetrical stages. Fig. 1 shows clearly the type of machine considered in this article.

The symmetrical stage lends itself quite readily to analysis. The problem here considered is: Given a reaction stage of specific physical dimensions, the rotational speed and the enthalpy change[1]—what is the internal work of the stage per pound of motive fluid? Of course, the converse of this problem involving design of the stage may be similarly treated, but this discussion will be restricted to the performance of a given stage.

First consider the turbine operating on steam. With a fixed amount of energy available to the given stage, determined by the enthalpy change from the initial state point of the steam on the Mollier diagram to the pressure at inlet to the succeeding stage, there are but two places that the heat released in the expansion may appear: (1) as mechanical work on the blading; (2) as reheat in the steam at the constant back pressure of the stage. The ratio of the mechanical work done on the blading to the enthalpy change is called stage efficiency and it is, of course, desirable to have this ratio as near unity as possible. The discussion of this point may be facilitated by introducing the concept of the condition curve (see Fig. 2).

The condition curve may be defined as the locus on the Mollier diagram of the state points of the steam at entrance to each of the stages in the turbine. Referring now to Fig. 2, it is clear that the slope of the portion of the condition curve representing a given stage is a measure of the efficiency of that stage. A condition curve parallel to the axis of entropy represents a perfect throttling, whereas one parallel to the axis of enthalpy represents a perfect conversion of heat to mechanical energy. These are, of course, limiting cases, neither of which ever actually occurs, but they help to clarify the picture.

The relationships between stage efficiency and various other terms may be expressed quite simply for the symmetrical stage. The conventional velocity triangles, shown in Fig. 3, are congruent. The following symbols and notation which have become fairly well standardized in this work are used here:

C_0 = Velocity equivalent of stage enthalpy change
C_{01}, C_{02}, etc. = Theoretical velocities
C_1, C_2, u_m, etc. = Actual velocities
α = Efflux angle
β = Angle of relative velocity

ν = Actual velocity ratio = $\frac{u_m}{C_1}$

ν_0 = Theoretical velocity ratio = $\frac{u_m}{C_0}\sqrt{2}$

δ = Mean cone angle of blading
φ = Blade velocity coefficient
η = Stage efficiency
H = Enthalpy change in the stage considered
W = Work done per pound of fluid on the blades in the stage considered

Any other symbols introduced will be explained.

$$\text{By definition } \eta = \frac{W}{H} \qquad (1)$$

[1] The use of the expression enthalpy change for heat drop at constant entropy is now quite general and hence will be adopted here.

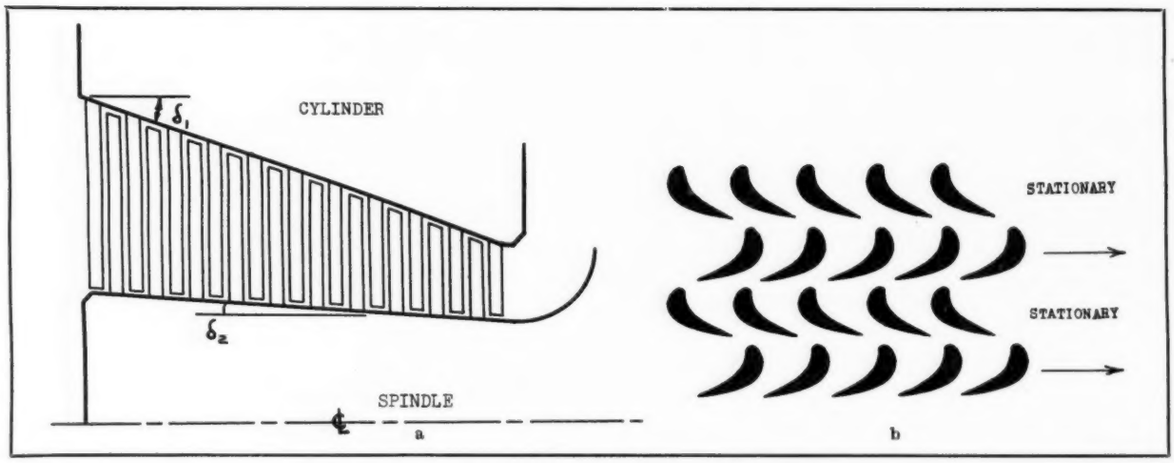

Fig. 1—Reaction turbine with symmetrical stages

a. Longitudinal section through spindle and cylinder
b. Cross-section of blades in two adjacent stages

Fig. 4 shows a portion of the blade annulus and the orientation of the velocity vectors with respect to the exit edge of the blade. Equating the impulse given the blades to the change in momentum of the fluid stream yields

$$T = \frac{r_m}{g}(2C_1 \cos\lambda \cos\alpha - u_m) \qquad (2)$$

where T is the torque per unit weight of fluid flowing per second and r_m is the mean radius of the stage.[2] The angle λ is generally unknown, but may be expressed in terms of the efflux angle α and cone angle δ. Thus from Fig. 4

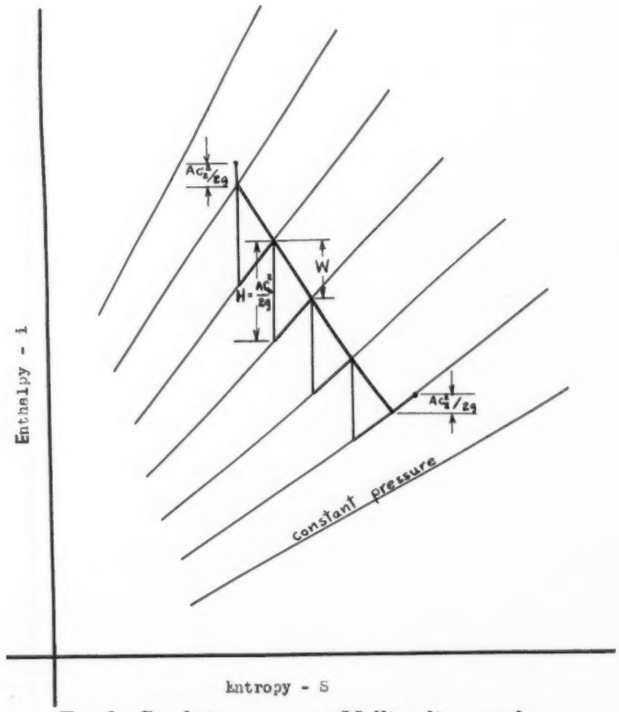

Fig. 2—Condition curve on Mollier diagram for a multistage turbine

[2] Equation (2) is strictly correct only when the ratio of blade height to mean diameter of the stage (h/M.D.) approaches zero. The use of symmetrical stages is arbitrarily limited to that portion of the blade path where h/M.D. ≤ .2. In this region the work expressions used in this paper are sufficiently accurate for nearly all purposes. Where values of h/M.D. exceed 0.2 special blades of non-uniform cross-section are used.

$$\tan\lambda = \sin\alpha \tan\delta \qquad (3)$$

The expression for work done on the blades per pound of fluid may therefore be written

$$W = T\omega = \frac{u_m}{g}\left\{\frac{2C_1 \cos\alpha}{\sqrt{1 + \sin^2\alpha \tan^2\delta}} - u_m\right\} \qquad (4)$$

For use in expressions to be derived later it is convenient to define a quantity ϵ, called diagram ratio, as follows:

$$\epsilon = \frac{gW}{C_1^2} = \frac{u_m}{C_1^2}\left\{\frac{2C_1 \cos\alpha}{\sqrt{1 + \sin^2\alpha \tan^2\delta}} - u_m\right\}$$

and since by definition $\frac{u_m}{C_1} = \nu$, the actual velocity ratio

$$\epsilon = \nu\left\{\frac{2\cos\alpha}{\sqrt{1 + \sin^2\alpha \tan^2\delta}} - \nu\right\} \qquad (5)$$

It is clear from (5) that ϵ is a function of ν only for a given group of blading.

In the foregoing expressions the work recovery has been derived from the actual momenta of the fluid leaving the blade passages. In calculating performance, however, only theoretical quantities are known. There

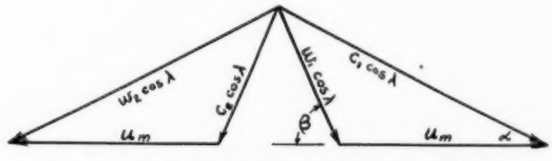

Fig. 3—Velocity triangles for symmetrical stage

must be established experimentally, a factor connecting the actual and theoretical cases, and expressing the fundamental losses associated with the flow of fluid through the blades. This factor is called velocity coefficient and is defined as the ratio of actual mean momentum per unit mass to theoretical momentum per unit mass for a given passage. It is usually designated by φ.

Introducing the velocity coefficient,

$$C_1 = \varphi C_{01} = \varphi\sqrt{\frac{C_0^2}{2} + C_3^2} \qquad (6)$$

Solving for C_0^2

$$C_0^2 = 2\left\{\frac{C_1^2}{\varphi^2} - C_2^2\right\} \qquad (7)$$

Also from the velocity triangles in Fig. 3 it is clear from the law of cosines that

$$C_2^2 \cos^2\lambda = C_1^2 \cos^2\lambda + u_m^2 - 2u_m C_1 \cos\lambda \cos\alpha$$

from which

$$\frac{C_2^2}{C_1^2} = 1 - \frac{\epsilon}{\cos^2\lambda} \qquad (8)$$

Substituting from (4), (5), (7), (8) and (3) in turn into expression (1) yields for stage, efficiency

$$\eta = \frac{W}{H} = \frac{\frac{\epsilon C_1^2}{g}}{\frac{C_0^2}{2g}} = 2\epsilon\frac{C_1^2}{C_0^2} = \frac{\epsilon C_1^2}{\frac{C_1^2}{\varphi^2} - C_2^2}$$

$$= \frac{\epsilon}{\frac{1}{\varphi^2} - 1 + \frac{\epsilon}{\cos^2\lambda}} = \frac{\epsilon}{\frac{1}{\varphi^2} - 1 + \epsilon(1 + \sin^2\alpha\tan^2\delta)} \qquad (9)$$

The stage efficiency at a given cone angle is thus expressed as a function of actual velocity ratio only if the quantities φ^2 and α are known. For a complete analysis there remains yet the relationship between the two velocity ratios. From Fig. 3

$$\tan\beta = \frac{C_1\cos\lambda\sin\alpha}{C_1\cos\lambda\cos\alpha - u_m} = \frac{\sin\alpha}{\cos\alpha - \frac{\nu}{\cos\lambda}}$$

which when solved for ν gives

$$\nu = \cos\lambda\left[\cos\alpha - \frac{\sin\alpha}{\tan\beta}\right] = \frac{\cos\alpha - \frac{\sin\alpha}{\tan\beta}}{\sqrt{1 + \sin^2\alpha\tan^2\delta}} \qquad (10)$$

Substituting from expressions (7), (8) and (9) the theoretical velocity ratio becomes

$$\nu_0 = \frac{u_m}{C_0}\sqrt{2} = \frac{u_m}{\sqrt{\frac{C_1^2}{\varphi^2} - C_2^2}} = \frac{u_m}{C_1}\frac{1}{\sqrt{\frac{1}{\varphi^2} - \frac{C_2^2}{C_1^2}}} =$$

$$\frac{\nu}{\sqrt{\frac{1}{\varphi^2} - 1 + \frac{\epsilon}{\cos^2\lambda}}}$$

$$\nu_0 = \frac{\nu}{\sqrt{\frac{\epsilon}{\eta}}} \qquad (11)$$

Having established eleven equations relating the variables involved in a consideration of the symmetrical stage, it is now possible to discuss the subject of stage efficiency from model tests profitably. First as to the need for model tests. The expense involved in the manufacture and test of full size, experimental, steam-driven stages is prohibitive. Furthermore, there is need for more detailed information than may be obtained by overall measurements of output and input. In experimenting on models, using air as a testing medium the observers may work with instruments directly in exposed jets without personal danger. More precise measurements may be made with air and the design and construction of the test equipment is simpler. These reasons are more than adequate to justify model testing with air.

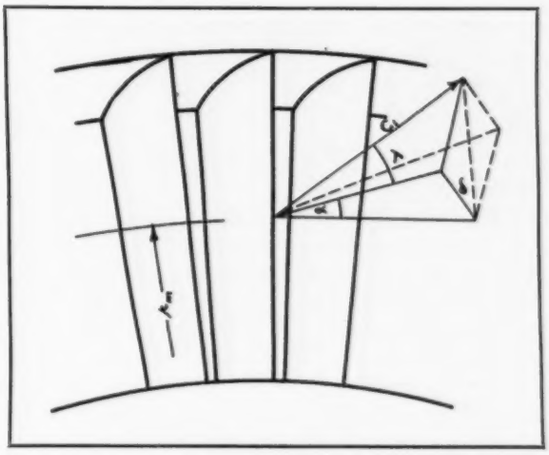

Fig. 4—Velocity direction for a blade mounted on cone with mean angle δ

Note—The angle α is the inclination of the plane of efflux with respect to the exit plane of the blades. The angle λ is the deviation of the maximum velocity from a normal to the exit edge of the blade in the plane of efflux.

From equation (10) it is clear that the actual velocity ratio may be expressed in term of the cone angle and the angles from the velocity diagrams only. Also from equation (9) the stage efficiency is determined for any case for which φ^2 and α may be measured, the cone angle δ, of course, being known. Remembering also that the stage is symmetrical, it follows immediately that if a properly designed model of several blades from a single row be mounted statically in an air stream at angles of attack β and provision be made for measuring φ^2 and α, the general problem of stage efficiency can be solved. Before going into the details of how this can be done it will be well to outline specifically the desired form for the final results, and keep this picture always in mind.

For use in calculating the work delivered by a group of symmetrical stages operating at various steam flows the test results of interest are: the maximum value of stage efficiency and a curve showing the percentage decrease in stage efficiency as a function of theoretical velocity ratio. Such a curve is shown in Fig. 5. The maximum efficiency occurs at a velocity ratio corresponding to the flow with least energy dissipation around the given blade section. The most nearly perfect flow conditions for an annular grid with foils of the type herein considered are attained when the fluid enters without shock or with slight negative shock. The meaning of the term shock is made clear by Fig. 6 which shows the orientation of the fluid at inlet to a given blade section at low, normal and high velocity ratios.

There are, in general, two well known methods of determining the velocity coefficient φ statically, namely, the impact traverse test, and the reaction test. Each method has some advantages, but because of a limitation which will later be explained, the reaction test is found inadequate. The impact test will, therefore, be considered as the means of obtaining the velocity coefficient.

Suppose, for example, that a model blade pack with three or more passages be mounted so that air enters at an angle β as shown on Fig. 7. The geometrical inlet angle of the blade is σ which leaves a large positive shock angle τ so that the condition represented simulates a very low velocity ratio. With an impact tube (usually called Pitot tube) small in proportion to the blade

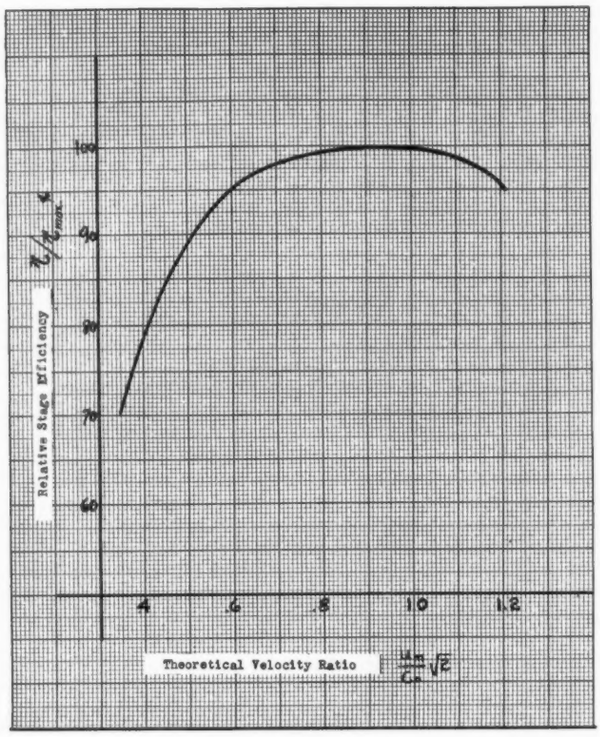

Fig. 5—Variation in efficiency of symmetrical stage with theoretical velocity ratio

passage dimensions, velocity profiles may be obtained for the center passage at various positions along its height. The possibility of establishing velocity profiles rests on the assumption that compression at the muzzle of the impact tube is isentropic. Because of so-called compression shock losses this assumption becomes invalid for supersonic velocities, but for the range of conditions encountered in a consideration of symmetrical stages, it is very probable that the compression *is* isentropic. A discussion of the type of apparatus and technique employed in obtaining accurate velocity traverses of passages as small as those considered here would be a separate article in itself, but it is sufficient to say that equipment has been made for controlling and measuring the position of the impact tube in space accurate to within $^1/_{1000}$ of an inch. Using this equipment, it is possible to obtain data for constructing a diagram similar to Fig. 8 by measuring the variation in impact pressure across the passage for a flow with the supply pressure P held constant. This diagram represents the conditions prevailing in a plane at a known position along the passage height. Similar diagrams may be obtained at other heights. Using a planimeter, the diagrams for the passage may be integrated tangentially and radially. It is only necessary to obtain the quotient

$$\frac{\int_{x_1}^{x_2}\int_{y_1}^{y_2} C^2\,dx\,dy}{C_{01}\int_{x_1}^{x_2}\int_{y_1}^{y_2} C\,dx\,dy}$$

which expressed in words is the ratio of actual mean momentum per unit mass[3] to theoretical momentum per unit mass or just the velocity coefficient φ of the passage.

It will be observed that in model testing at conditions corresponding to a low velocity ratio, the velocity at inlet to the test pack is high. A considerable part of the energy release occurs in the entrance passage and is therefore affected by the coefficient of this passage. It is clear, therefore, that any method of arriving at a coefficient for the blades alone which depends upon measurements made on the whole pack must be in error by an amount depending upon the coefficient of the entrance passage and the percentage of the stage drop occurring therein. This is the reason why a reaction test for the determination of the blade velocity coefficients is inadequate except for a very limited range of velocity ratios. The following paragraph will amplify this point.

In a reaction tester, the blade pack and entrance passage are mounted so that the force resulting from the escape of a measured mass of fluid per second may be determined in magnitude and direction. The quotient of the force by the mass flow per second gives directly the mean actual velocity of escape. Initial pressure and temperature readings establish the theoretical velocity of escape and the ratio of actual to theoretical efflux velocity is the velocity coefficient. Since, however, these results are an average for the whole model pack they contain the characteristics of the entrance passage which at the lower velocity ratios is sufficient to completely vitiate the results. From the commercial and economic standpoint the region of low velocity ratios is the interesting part of the stage efficiency diagram since the lower the velocity ratio commensurate with the desired efficiency, the fewer the number of stages necessary to absorb a given enthalpy change. The need for a more general method of measurement of velocity coefficient accurate over the entire operating range of velocity ratio is thus a very real one. The solution is provided by the

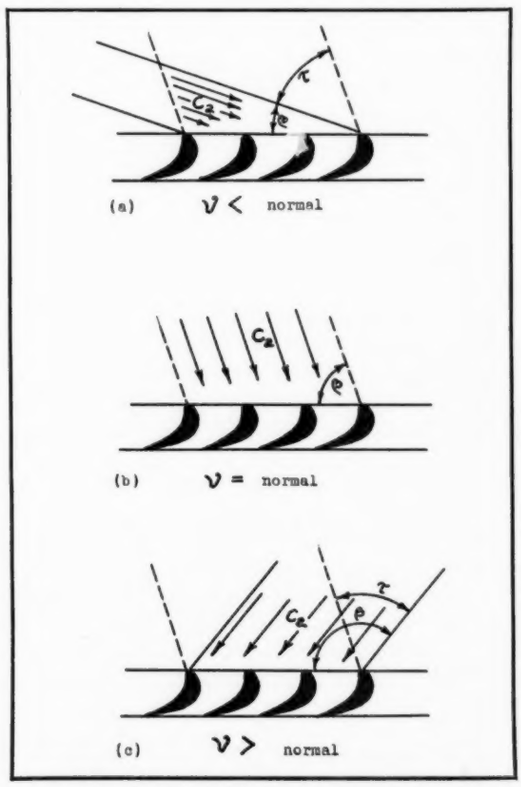

Fig. 6—Static attack angles associated with various velocity ratios

[3] The effect of density variation in the plane of efflux is negligible.

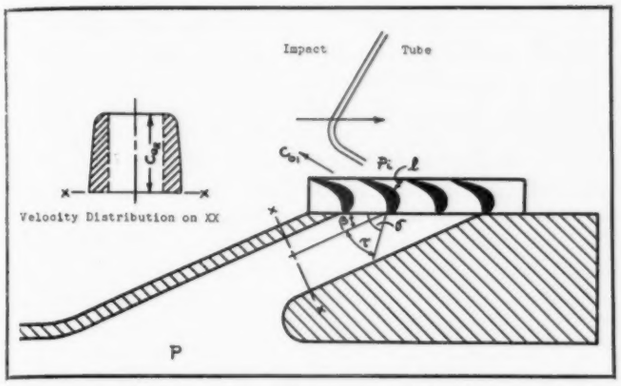

Fig. 7—Elevation of model arranged for impact test

impact traverse test already described. That the accuracy of this type of test is not impaired by conditions in the approach passage will now be made clear.

When a passage flows full, without loss of fluid contact with the walls, the only energy loss is friction in the boundary layer of fluid adjacent to the passage wall. The characteristic turbulent velocity distribution for an approach passage of the type used in model tests of reaction blades is shown in Fig. 7. Across more than 90 per cent of the width, the velocity is quite uniform and equal to the theoretical maximum. Surrounding this core of uniform velocity is a thin shell in which the velocity drops sharply to zero at the passage wall. This thin shell is known as the boundary layer, and it is the stability of this layer that dictates whether the flow in an entire passage shall be relatively efficient or inefficient.

In a case of flow accompanied by a falling pressure, as in a nozzle, the pressure gradient in the boundary layer provides the motivating force on the fluid particles to overcome the friction drag of the wall and there is no tendency toward stagnation. The viscous drag transmitted from the more rapidly moving particles in the center is in the same direction as the pressure drop. The boundary layer is then thin and stable, and there is no tendency for the flow to leave the wall. In a case of flow accompanied by a rising pressure, as in a diffuser, there is no motivating force in the boundary layer in the direction of flow other than that transmitted by viscous drag from the main body of the stream. Wall friction brings the outside layer of particles to rest and a local reverse pressure gradient is established. This reverse pressure gradient forms a couple with the viscous drag forces transmitted to the particles in the boundary layer from the main body of the stream. The boundary layer becomes unstable and leaves the wall. Vortices are formed diverting primary energy of the stream into

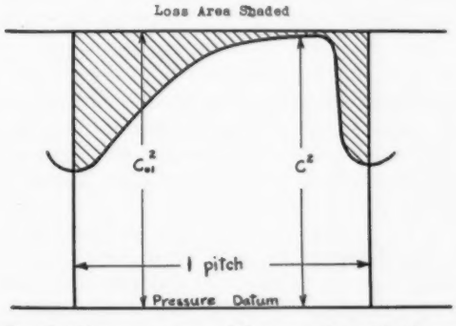

Fig. 8—Traverse record for reaction blade pack

secondary flows. These conditions are, of course, destructive to the velocity distribution intended.

The approach passages for the model tests are designed to avoid a loss of contact with a flow accompanied by a falling pressure. For approach angles corresponding to any velocity ratio, it is certain, therefore, that the center passage of the model blade pack is unaffected by any losses associated with the entrance conditions. A traverse test of the center passage therefore measures only losses associated with the flow in the blade and hence yields a true measurement of the blade velocity coefficient.

It has been pointed out that a measurement of the efflux angle α is also a necessary part of the static test. One very simple and fairly accurate way of doing this is to align, by eye, a straight edge held in a bevel protractor with a fine silk thread, connected to a small, thin rod and held in the jet. The edge of the protractor is held parallel to the exit plane of the blades, and the angle may be read directly. Small, light metal vanes of various shapes, supported on a tightly stretched piano wire in the jet have been used in place of the thread, but without marked success. The chief difficulty with a method of this kind is that the indicator flutters to a certain extent, and it is not possible to accurately align a measuring device with it.

An alternative, though rather elaborate, method for efflux angle determination is the two-component balance. The test pack is mounted in a frame containing two independent balances at right angles. Means must be pro-

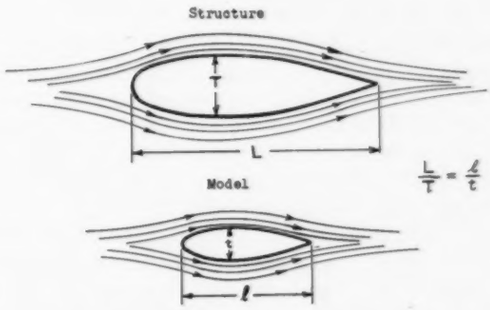

Fig. 9—Mechanically similar flows

vided for supplying air to the pack without exerting any forces on the balances. If this can be done satisfactorily, the balances will indicate only the two components of force resulting from the escape of the jet. The efflux angle α follows immediately from the tangent relationship. As may be easily guessed, the difficulty in this method lies in conducting air to the pack without exerting forces on the balances. A fairly satisfactory way of doing this has been devised, and using the necessary calibrations, angle measurements accurate to within about 0.5 deg may be obtained. Comparatively large errors in the efflux angle scarcely affect the stage efficiency η, but they do affect the velocity ratio ν_0, and the possible error increases with the deviation of the velocity ratio from the peak point of the diagram. It is freely admitted that up to the present time the measurement of the angle α is performed with considerably less finesse than characterizes the measurement of the velocity coefficient φ. On the other hand, tests of the kind described afford the only means of direct measurement of the efflux

angle. Turbine tests can yield this quantity only indirectly with certain assumptions, and the results must then also be provisional.

Methods of obtaining velocity coefficient and efflux angle from blades statically mounted in an air stream at given angles of attack have now been discussed and, as has been shown, these quantities are sufficient for a general evaluation of stage efficiency. One important question remains: What constitutes a properly designed model? The answer to this involves a part of the general theory of mechanical similarity, but because passages of the same range of size are used in both air model and steam turbine the problem is greatly simplified.

Fluid flows through passages or around obstructions are said to be mechanically similar when the configurations of the streamlines at corresponding points are similar. Similarity of the streamlines means that the direction of the velocity at corresponding points is the same. A classic example of mechanically similar flows is the case of the simple airfoils in Fig. 9. When mechanical similarity exists, all dimensionless ratios, such as φ, for model and prototype are interchangeable without correction.

Obvious from the definition is the conclusion that mechanical similarity presupposes geometrical similarity, since like streamline patterns around dissimilar bodies are impossible, but this condition though necessary is not sufficient. The ratio of forces on a fluid particle at a point determines velocity direction at that point. For mechanical similarity then, the ratio of forces on fluid particles at points similarly situated with respect to geometrically similar bodies must be the same.

Restricting the field to the case of a medium with no free surfaces, the forces on the fluid particle to be considered arise from inertia, viscosity and elasticity. Gravity is eliminated by aerostatic lift since the particle is a totally immersed body. In a study of the flow of compressible fluids such as air or steam, elastic forces must be considered only when pressure differences are great enough to cause appreciable change in density. The velocities associated with appreciable density changes are quite high (near sound velocity) and it may be safely stated that elastic forces have only a minor influence on the streamline pattern for the range of flow conditions encountered in the symmetrical stage.

There remains then only the influences of inertia and viscosity, and the shape of the streamlines in a particular case will depend principally upon the ratio of inertia forces to viscosity forces. This ratio called "Reynolds Number," in honor of the English scientist who first announced its significance, is a dimensionless quantity which may be expressed in terms of a characteristic length, a characteristic velocity and the kinematic viscosity of the fluid. Thus in Fig. 7 the Reynolds Number for the flow through the blade pack is $R = C_{01}l/K$ where K is the kinematic viscosity of the fluid at the blade exit. Kinematic viscosity is simply absolute viscosity, i.e., the shearing force per unit area per unit velocity gradient, divided by the mass density. In the case of the reaction blades it would be permissible to use a dimension other than "l" for the characteristic length and a velocity other than C_{01} for the characteristic velocity and hence obtain some other number that could be compared with another case similarly treated, but the use of these quantities has become customary.

In symmetrical stages, equality in the Reynolds Numbers for several cases of fluid flow through geometrically similar passages indicates mechanical similarity. Equality in the Reynolds numbers gives assurance that dimensionless ratios pertaining to the several cases are the same. Inequality in the Reynolds numbers, however, does not necessarily mean that mechanical similarity does not exist. There are any number of instances where the streamline pattern throughout a wide range of Reynolds numbers is practically unchanged. It is, therefore, not only necessary to know the law on which similarity depends but also how it depends upon that particular law. For some of the stage efficiency studies it is possible to test the model at a Reynolds number differing widely from that for the prototype and yet draw accurate conclusions. This is the main reason why model tests are so universally applicable in this range of flow conditions.

With tests at velocity ratios differing widely from that for which the section was designed the flow in the blade passage is highly disturbed by eddies at the ends resulting from a loss of fluid contact. Under these conditions the velocity coefficient is particularly sensitive to small deviations from geometrical similarity. This point must be borne in mind in comparing results, especially those obtained at low velocity ratios.

Model construction and testing, in general, involves a compromise. It is an unusual case where all the conditions prerequisite to mechanical similarity can be satisfied. The need arises for a decision, therefore, in almost every case, as to what is the dominant factor in establishing mechanical similarity. Extended experience alone furnishes the knowledge necessary in deciding wisely. Model tests may be made quickly, accurately and inexpensively, but in conclusion it seems appropriate to add that a background of many observations is not a small factor in obtaining reliable results.

Robert H. Wyld has recently become associated with Burns & Roe, power plant consultants at 233 Broadway, New York. Previous to this connection Mr. Wyld was for a number of years with the Foster-Wheeler Corporation.

Earle W. McMullen has been made Director of Research of the Eagle-Picher Company, succeeding Dr. John A. Shaeffer who resigned to become president of Franklin and Marshall College.

J. E. N. Hume, formerly Assistant Manager of the General Electric Company's industrial department, has been appointed manager of that department, succeeding the late W. W. Miller.

Alton Kirkpatrick, for a number of years with Albert C. Wood, well known consulting engineer of Philadelphia has joined the latter under the firm name of Wood & Kirkpatrick with offices at 1411 Walnut Street, Philadelphia, Pa.

W. E. S. Dyer, formerly consulting engineer of Philadelphia and more recently President of the Edge Moor Iron Company, has opened a consulting engineering office in New York.

Effect of Type of Surface on Radiation Loss from Furnaces

By **B. J. CROSS**

Combustion Engineering Company Inc.

A brief discussion of the absorption of heat by furnace walls, its transfer through the wall by conduction and its dissipation by radiation and convection. Basic formulas and curves for determining the heat loss are included and the effect of different emissivity factors for various surfaces is commented upon.

THE so-called radiation loss from furnaces involves all three of the modes of heat transfer. The flames and incandescent surface of the fuel bed radiate heat to the inner furnace walls; this heat flows through the walls by conduction and is lost from the outer surfaces by radiation and convection. In order that there may be a flow of heat, temperature differences must, of course, exist. The temperature of the furnace must be higher than that of the inner furnace walls; there must be a temperature drop across the refractories of the wall and the outer surfaces of the furnace walls must be at a higher temperature than the surrounding air and its inclosing surfaces. In a condition of equilibrium the quantity of heat flowing through the various sections of the path is constant and the temperature drops along the path of flow are fixed by the conditions for the particular mode of heat transfer involved. The expressions equating this constant amount of heat over the different portions of its path would be as follows, unit area and unit time being assumed.

$$KE_1 (T_f^4 - T_w^4) = \frac{k}{l} (t_w - t_s) = KE_2 (T_s^4 - T_r^4) + C (t_s - t_r)^{5/4}$$

Where, K = Radiation constant
T_f and t_f = Temperature in furnace
T_w and t_w = Temperature of inside furnace wall
T_s and t_s = Temperature of outside furnace wall
T_r and t_r = Temperature of room
E_1 and E_2 = Emissivity factors
C = Convection constant
k = Conductivity of wall material
l = Thickness of wall material—inches
Fahrenheit scale temperatures are indicated by t and absolute temperature (°F + 460) by T.

The diagram of Fig. 1 shows a typical temperature gradient over the path of flow represented by these equations.

The furnace temperature t_f may be considered to be fixed by conditions for efficient operation of the furnace. It can be reduced by admitting a higher percentage of excess air than that required for efficient combustion though this is usually undesirable. It can also be lowered by increasing the useful absorption of heat in the furnace as by installing water-cooled walls connected into the boiler circulation.

The transfer of heat from the flame to the furnace walls is almost entirely by radiation and is proportional to the difference between the fourth powers of the absolute temperatures of the flame and the wall. The usual form of the Stephan-Boltzman law expresses this relation as

$$Q = 1720 \left[\left(\frac{T_f}{1000} \right)^4 - \left(\frac{T_w}{1000} \right)^4 \right] E_1$$

Where Q is the heat transmitted in Btu per sq ft per hr. This may be written

$$Q/E = 1720 \left(\frac{T_f}{1000} \right)^4 - 1720 \left(\frac{T_w}{1000} \right)^4$$

Therefore, if we evaluate the expression $Q/E = 1720 \left(\frac{T}{1000} \right)^4$ for each value of T, the net transfer of heat may be determined by subtracting the lesser value from the greater and multiplying the difference by the value for emissivity, E.

The curve of Fig. 2 gives values for $1720 \left(\frac{T}{1000} \right)^4$ plotted against temperature. A portion of this curve is shown in large scale. It will be seen that at usual furnace temperatures only a small temperature difference is necessary to transmit heat at the rate of the order of losses through walls. A 5-deg temperature difference will account for a transfer rate of almost 1000 Btu per sq ft per hr. This holds for a clean wall only. Actually the wall may have a slag deposit and a greater temperature difference may exist for this transfer rate. The change in wall temperature, however, for a difference in heat transfer rate of less than 100 Btu will be negligible and for the purpose of this article, the temperature of the inside furnace wall will be considered as constant.

The rate of heat transfer through the wall by conduction is proportional to the conductivity of the material, to the temperature difference of the two sides and to the wall thickness. The conductivity is the amount of heat transferred through a unit area in unit time, for a one-degree temperature difference across a unit thickness, or

$$k = \frac{\text{Btu}}{\text{Sq ft} \times \text{hr} \times \text{°F/in.}}$$

Conductivity is a property of the material. Conductance, H, is a property of the wall or layer in a wall and is equal to conductivity divided by thickness in inches, or

$$H = \frac{k}{l}$$

When a wall is made up of different materials, the overall conductance H_0 of the wall may be determined from

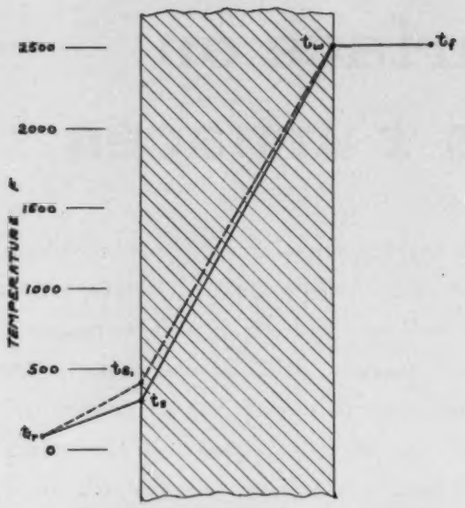

Fig. 1—Diagram of temperature drops for refractory wall

the individual conductances of the different layers by the relation:

$$\frac{1}{H_0} = \frac{1}{H_1} + \frac{1}{H_2} + \frac{1}{H_3}$$

Thermal resistance R is defined as $\frac{1}{H}$. The thermal resistance for a composite wall is, therefore, equal to the sum of the thermal resistance of the parts;

$$R_0 = R_1 + R_2 + R_3$$

The quantity of heat transmitted through a wall, in Btu per square foot per hour is equal to the temperature difference times the conductance, or by the temperature difference divided by the thermal resistance.

The temperature gradient across a wall may, therefore, be represented by a straight line, as in Fig. 1, for either a solid wall of one material or for a wall made up of different materials. This line indicates the overall temperature drop but does not represent temperatures within the wall.

The conductivity of refractories will vary over a considerable range. It is inversely proportional to porosity and will, therefore, vary with the age of the refractory as the porosity decreases. It also will vary directly with temperature.

Fig. 5 gives a set of values of k for refractory firebrick, (Norton's values) and for insulating firebrick, the latter values being taken from *Technical Data on Fuel* by H. M. Spiers.

The loss of heat from the outer surface of a wall takes place by both radiation and convection. The transfer of heat by convection is proportional to the five-fourths power of the temperature difference between the surface and the room air. It is affected to some extent by the type of surface, being greater for smooth than for rough surfaces. It is also considerably affected by the position of the surface. It is greater for a horizontal surface facing upward than for a vertical surface, and greater for a vertical surface than for a horizontal surface facing downward. The expression for the heat lost by convection is

$$Q_c = C \times 0.32\ (t_s - t_r)^{5/4}$$

Where Q_c is in Btu per square foot per hour, C is a constant for position of surface and may be taken as 1 for vertical surfaces. For upward horizontal surfaces C may be 1.2 and for downward facing horizontal surfaces a value of 0.8 may be taken. A considerable latitude for individual preference is permitted in the choice of values for this constant.

The rate of heat transfer by radiation from the outer surface of a wall is evaluated from the Stephan-Boltzman

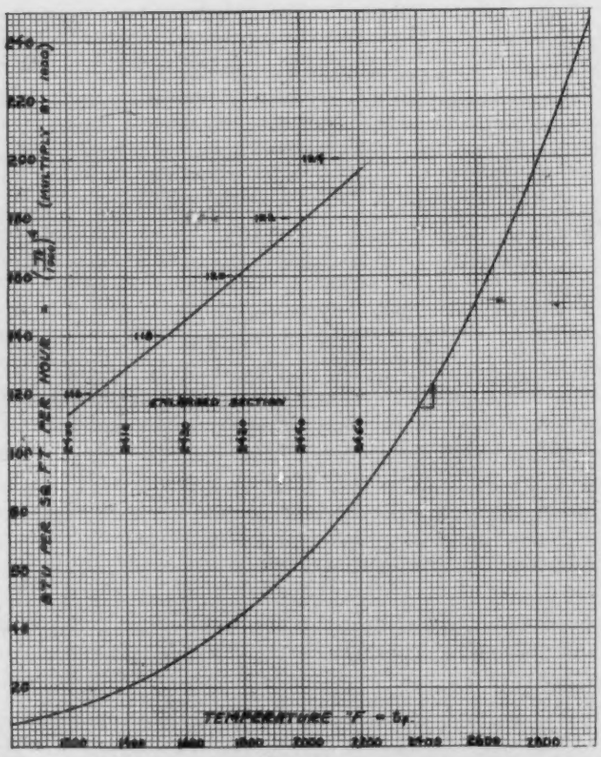

Fig. 2—Radiation rates—high temperature range. (Enlarged section is that indicated by small triangle.)

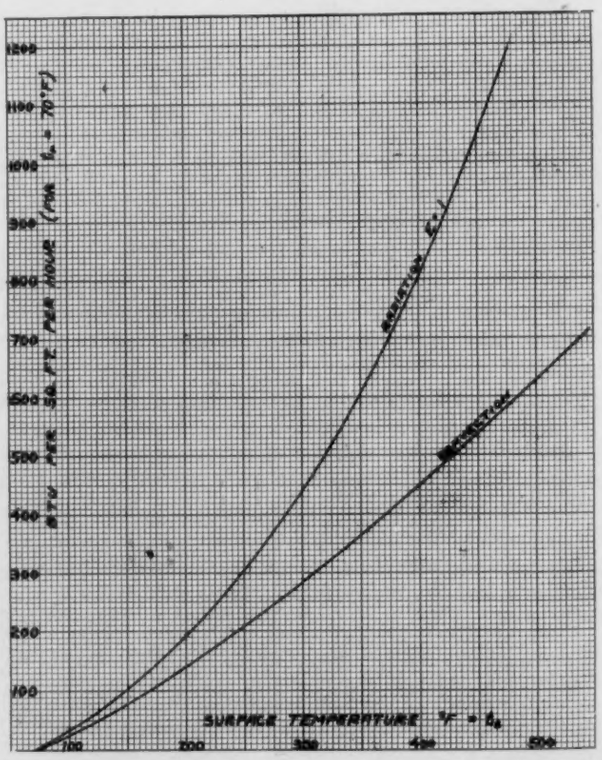

Fig. 3—Radiation and convection losses from surface of wall to surroundings at 70 F.

law in the same manner as for the radiation from the flame to the inner surface of the wall. The value of E, the emissivity factor, which for flame radiation is about unity, will, for the outer wall surface, vary considerably with the type of surface. For brick and also for dark painted steel or iron surfaces, this factor is 0.9 or higher. For aluminum painted surfaces, which are now quite common in boiler rooms, the emissivity may be as low as 0.25, though there will be some variation with the quality of the paint and with its condition. For average conditions the emissivity of aluminum painted surfaces will vary between 0.3 and 0.5. The curves of Fig. 3 show the radiation and convection rates plotted against surface temperature. A room temperature of 70 deg is assumed.

If the temperature of the surface would remain constant, the application of aluminum paint would reduce the radiation component of the surface loss in proportion to the change in emissivity. However, because there is a reduction in radiation with a change in emissivity, the surface temperature will rise as less heat is taken away. The entire gradient through the wall will change until a new equilibrium point is reached where the heat transmitted through the wall becomes equal to that lost from the surface. Thus, in Fig. 1, the solid line represents an equilibrium condition for a wall having an outer surface emissivity of 0.93. When the emissivity is changed to 0.3, the wall temperature will rise though the heat lost will be less. The gradient across the refractory will also change and the temperature of the inner wall will increase. This latter change is small and may be neglected. The new equilibrium condition is indicated by the broken line.

The curves of Fig. 4 have been prepared to offer a graphical method for the determination of equilibrium conditions resulting from a change of the emissivity factor of the outer wall surface. The combined radiation and convection losses are shown for several radiation emissivity values and varying surface temperatures. Curves showing the conduction transfer rates for several types of walls are plotted across the surface loss curves. The equilibrium conditions are given by the intersection

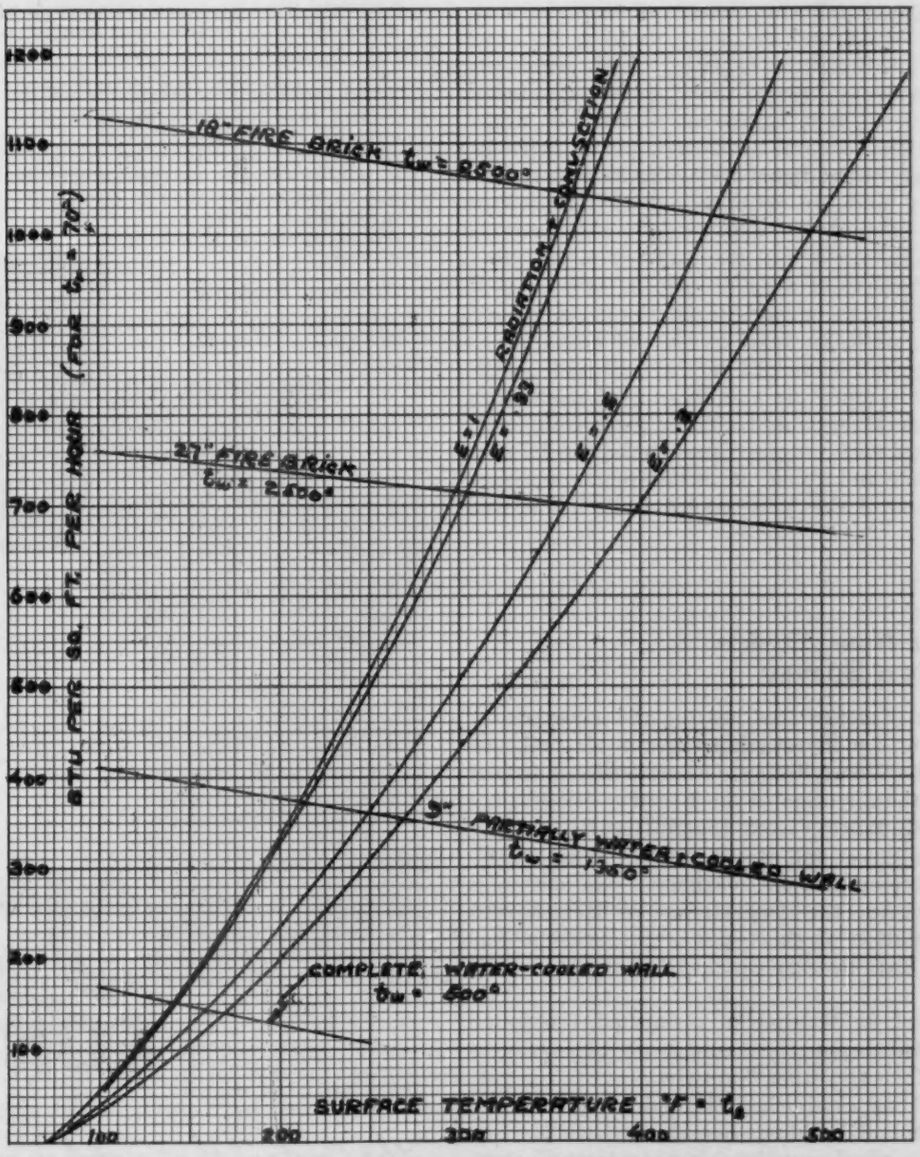

Fig. 4—Curves for determining equilibrium conditions for furnace walls with outside surfaces of different emissivity

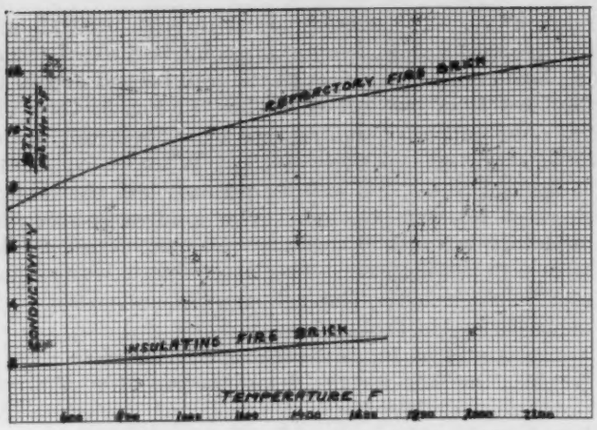

Fig. 5—Heat conductivity values for refractory firebrick and insulating brick

of the conduction transfer lines with the surface loss lines. Thus, for a 27-in. firebrick wall with an inner surface temperature of 2500 F and an outer surface emissivity of 0.93, the equilibrium temperature of the outer surface will be 305 F and the heat loss from the outer surface will be 710 Btu per sq ft per hr. If the emissivity of the outer surface is changed to 0.3, a new equilibrium condition will occur at a surface temperature of almost 400 F and the heat loss will be 690 Btu per sq ft per hr.

The loss curves of this figure are determined by surface conditions only and are proportional to temperature difference and emissivity. Wall transfer rates may be calculated for any type of wall and drawn in on this sheet. Curves for four types of walls are shown, two for solid refractory walls of 18 in. and 27 in. thickness, respectively, one for a partially water-cooled wall having tubes on 9-in. centers half imbedded in the wall and backed by 6 in. of conforming tile and $2^1/_2$ in. of moderately compressed mineral wool, and a curve for a completely water-cooled wall with tangent tubes backed up by 3 in. of conforming tile and $2^1/_2$ in. of moderately compressed slag wool. Each of the water-cooled walls has a steel casing.

Progress of New York's Air Pollution Survey

An inspection of all furnaces, heating and power plants, and incinerators, by investigators of the air pollution survey which the Works Progress Administration is conducting under the direction of the Board of Health, has been made in Manhattan from the Battery to as far as Ninety-Seventh Street and if the present rate is maintained will be completed in February. The survey is intended to provide the Department of Health with accurate information of the types of furnace, the fuel used and other data bearing upon the possibility of air pollution.

When the work in Manhattan is finished, the squad of inspectors will begin the same type of survey in Brooklyn and then the other boroughs.

The project has established thirty-five observation posts on the tops of buildings distributed over the five boroughs. This number will be increased until there is a total of fifty-three such posts.

"ALWAYS ON THE MOVE"

- - that's the A-Jacks

WATCH the uptake damper in a boiler furnace equipped with A-Jacks Combustion Control. While other controls would be standing still the A-Jacks is working—finding the right position to *keep* steam pressure or over-fire pressure right—hurrying on to the next position—always working—watching everything.

A good combustion control *must* do just that. Steam demands are fluctuating constantly—puffy winds and eddies are constantly affecting draft—only a highly sensitive, quick acting, powerful regulator can keep up with these ever changing conditions. Thousands of installations have proved that A-Jacks Combustion Controls can and do. As many operating records tell the story of this better control in plain terms of fuel saved.

The A-Jacks Damper Regulator and the A-Jacks over-fire control installed as shown below and made still more effective by the use of National Cam Valves, have a remarkable record in stoker fired plants. The A-Jacks is just as effective in smaller installations as in larger plants. Ask for complete information covering design, application, and performance.

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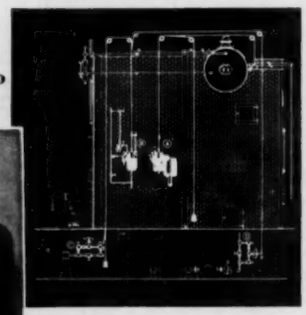
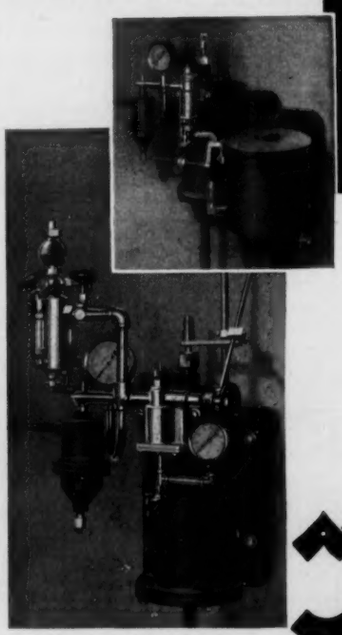




A-Jacks Over-Fire Control (opposite)—used to control up-take damper, compensating for over-fire pressures produced by varying coal feed or induced draft. A typical hook-up of the A-Jacks Regulator and Over-Fire Control is shown in the diagram above. This is merely one of many effective hook-ups.

A-Jacks Damper Regulator—used in connection with highly accurate National Cam Valves to proportion stoker and blower speeds to steam demand. Operated on boiler pressure.

A-JACKS CONTROL

"the Control that watches everything"

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Hot-Water Accumulator on District Heating System

In a recent paper before the Institution of Heating and Ventilating Engineers, A. Margolis describes the Hamburg district heating system which is supplied largely with steam from back-pressure turbines that supply electricity to the city. Heat is furnished both by a steam distribution system and hot-water lines. In connection with the latter there is employed a large vertical hot-water accumulator of welded construction and resembling a gasometer. It is mounted in the open and serves both as an expansion tank and as a heat accumulator to assist in balancing the diversity between power and heating demands. Its height varies from 65 to 130 ft and the capacity is 550,000 gallons. Thermal losses from the accumulator are said to be very small and the drop in water temperature over 24 hr is less than one degree. The paper is reviewed at length in the December issue of *The Steam Engineer* (London).

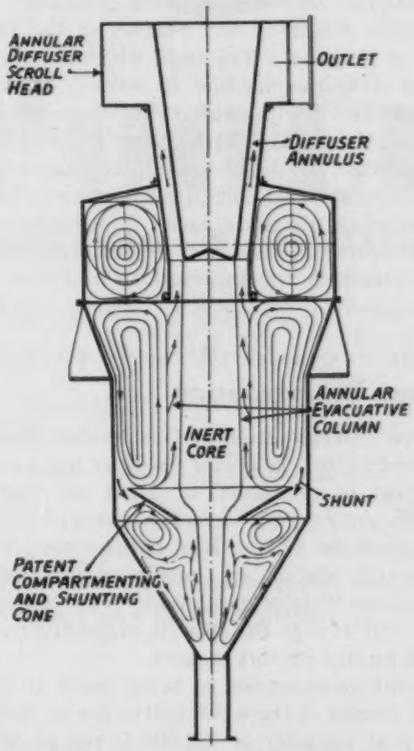
Circulation in Cross-Drum Boilers

Die Wärme of November 2 contains a discussion by A. Doebler of circulation in cross-drum sectional-header water-tube boilers. Pointing out that circulation in the upper tubes may be reversed under certain load conditions, the author advocates steeper inclination of the tubes and a limited number of tubes per header section, also multiple connections between the headers and the steam drum. He cautions against the introduction of throttling devices to control circulation in certain tubes, but does not regard the heating of the downcomers as objectionable. The Doebler boiler is described as embodying the design features advocated by the author.

A New Dust Precipitator

A new design of dust precipitator which is claimed to be an advance on the usual type of cyclone has been developed by the British firm of Fraser & Chalmers and is described in the December issue of *The Power and Works Engineer*. Cyclones depend for their action on the difference in density between the carrying medium (air or gas) and the solid matter or dust to be precipitated. It is pointed out that any attempt to increase the degree of separation of dust from gas by raising the inlet velocity intensifies a phenomenon by which the lower of the two vortex rings set up by centrifugal force reentrains part of the dust that is forced to the shell, carrying it upward with the gas to the discharge pipe and out of the cyclone.

The new precipitator which is called the "Micron," overcomes this entraining action by the addition of a compartmenting and shunting cone, which provides a lower boundary terminating the lower pair of vortex rings and allows a shunt current to pass to a secondary chamber below where a further pair of vortex rings is set up. Only a small portion of the gas passing into the precipitator reaches the lower chamber, with the result



Section through "Micron" precipitator showing compartmenting and shunting cone

that only an inappreciable quantity of dust is entrained and carried through the central opening in the cone to the evacuative column. This method of controlling the vortex rings is claimed to contribute greatly to the dust-separating efficiency.

Another feature of the design is the provision of an annular diffuser in the outlet pipe.

Efficiencies of 90 to 98 per cent are claimed, depending upon the density and the particle size.

Experience with British High-Pressure Station

The Electrical Review (British) of November 15 contains an abstract of the report of the Bradford Electricity Committee for twelve months' operation, through

October, in which reference is made to the 1100-lb installation in the Valley Road Power Station. The high-pressure unit ran for 1250 hr during the year. The early troubles due to blade corrosion in the turbine have apparently been overcome. Some deterioration of the reheater elements in the boiler have been noted, but the greatest difficulty has been in maintaining the refractory lining of the furnace, which resulted in frequent outages. There have been no difficulties, however, that are directly attributable to the high pressure.

An 80,000-Kva, 3000-Rpm Turbine-Generator

During the past year, according to *Die Wärme*, an 80,000-kva, 3000-rpm turbine-generator was installed at the Schelle Power Station of the Interscant Company in Belgium. This unit, which is claimed to be the largest 3000-rpm machine in service, has one high-pressure and two low-pressure cylinders in tandem on a single shaft the total length being 55 ft. The high-pressure element is made up of one impulse and 23 reaction stages and each low-pressure element has double flow through four reaction stages. The steam conditions are 515 lb pressure and 850 F temperature. Three stages of extraction are employed for feedwater heating.

Battersea Performance

Battersea Power Station of the London Power Company, according to *The Steam Engineer* has a record, for its first year of operation, of 27.24 per cent overall thermal efficiency on a load factor basis of 53.7 per cent. This is equal to 12,525 Btu per kw-hr. The coals burned in this station are mostly of the Scotch and semi-anthracite Welsh types having gross heating values of 11,000 and 13,500 Btu per lb, respectively, and are burned on multiple-retort stokers.

At present an extension is being made to Battersea which will consist of three 300,000 lb per hr boilers, each having a peak capacity of 400,000 lb per hr, which will furnish steam to a 110,000-kw turbine-generator.

Extension to Dalmarnock

The *Electrical Times* (British) states that the Dalmarnock Station, supplying Glasgow, is being extended by the addition of three 50,000-kw turbine-generators. One is now being installed, the second will go into service next year and the third will be ready in 1938. Three 200,000 lb per hr boilers have been ordered for the last-mentioned unit.

This station, it will be recalled, went into operation shortly after the close of the War and contained five 18,750-kw turbine-generators operating at 275 lb steam pressure and 900 F temperature. For several years it was one of the outstanding stations in Great Britain and maintained a station heat rate of 19,000–20,000 Btu per kw-hr which was excellent for that period and the prevailing steam conditions.

Specific Heats of Gases and Vapors at High Temperatures

Calorimetric determinations of the specific heat of gases at high temperatures offer unusual difficulties because the heat capacity of the very thin gases is small compared to that of the calorimeter.

To determine the specific heat of gases and vapors with much greater accuracy they have been calculated from spectroscopic measurements.

New values of mean specific heats of various gases encountered in combustion and including steam and air are given in *Archiv für Wärmewirtschaft & Dampfkesselwesen*, Vol. 16, No. 12, 1935, and are the results of determinations by E. Justi from spectroscopic measurements. They range from 0 to 3000 C, are given in large calories per Kg-mol and degrees and low constant pressures and are to be considered for $p = 0$ atm abs. An accompanying table prepared by J. Otto, but not published in *Wärmewirtschaft*, permits the conversion of these specific heats to pressures of about 20 atm abs. The corrections are, however, so small that the values of the tabulated specific heats may be used for $p = 1$ atm abs without material error.

The effect of dissociation, which has the effect of increasing the specific heat at the higher temperatures, is disclosed in a table of corrections.

A comparison of the most accurate old and the new methods of determining the specific heats is shown in several curves at high and low temperature ranges and justify the new method.

Like the calorimetric method the spectroscopic method is based on empirical determinations.

Large Extraction Turbine in Paper Mill

The December issue of *Brown Boveri Review* describes a 10,000-kw extraction turbine-generator set which it is building for the Steinhagen & Sänger Paper and Cellulose Works at Myszkow, Poland. This is the third such set that has been installed in this plant, the other two being of 2750 and 6500 kw capacity; the latest unit, however, is of particular interest because it is one of the biggest extraction turbines that has been built by Brown Boveri & Company. It is a two-cylinder machine, the high-pressure element having two rows of impulse blades followed by a drum of reaction blading and the low-pressure element a single row of impulse blades followed by a large number of rows of reaction blades. The rotative speed is 3000 rpm.

The initial steam conditions are 278 lb per sq in. and 750 F temperature. Maximum steam extraction is 66,000 lb per hr at 50 lb gage.

Coal Consumption of Different Countries

The following figures of coal consumption in the chief European countries and also Canada and the United States, for the years 1913, 1929 and 1934, were given in a

recent issue of *Glückauf* and reprinted in the December issue of *The Fuel Economist.*

Coal Consumption in Thousands of Tons

				Per Head of Population		
	1913	1929	1934	1913	1929	1934
Germany	180,050	170,573	133,428	2.69	2.67	2.04
Great Britain	192,127	176,285	164,346	4.17	3.96	3.61
Holland	....	13,166	12,841	1.72	1.69	1.54
Belgium	26,046	38,486	27,914	3.41	4.77	3.38
France	64,834	96,629	76,744	1.63	2.31	1.75
Spain	7,591	9,643	7,415	0.37	0.43	0.30
Italy	11,343	15,449	13,400	0.32	0.38	0.32
Czechoslovakia	....	29,013	18,311	..	1.99	1.19
Hungary	9,070	9,352	7,195	1.13	1.10	0.84
Poland	....	32,354	19,166	..	1.07	0.58
Sweden	5,763	6,671	7,102	1.02	1.09	1.15
Russia	44,644	40,136	89,971	0.26	0.27	0.57
Japan	18,115	35,613	36,036	0.34	0.56	0.53
Canada	29,539	33,631	26,015	3.81	3.44	2.39
United States	494,734	532,788	363,683	5.10	4.37	2.87

The increased consumption per head of population in France since 1913 is due to a great development of the heavy industries, such as iron and steel production, and also the inclusion in the French statistical area of the Saar industrial region. Russia also is a country which is rapidly developing industrially, which accounts for the increased consumption of coal per capita last year as compared with 1913. In the United States, the reduction in coal consumption per capita since 1913 can be reasonably supposed to be due largely to more extensive use of oil and natural gas fuels since 1913, and to some extent this is true also of Canada.

In countries such as Great Britain and Germany, the decline in coal consumption per capita since the war is scarcely accounted for by its replacement with oil fuel. In the meantime, industrial production has considerably increased which, all other things remaining equal, should lead to an increase of coal consumption. It must be assumed therefore, that coal is being utilized in industry with much greater efficiency than formerly.

World Power Conference Plans

Plans for the Third World Power Conference, to be held in Washington, D. C., September 7–12, 1936, are rapidly taking definite form. Unlike the previous conferences there will be no strictly technical papers but instead emphasis will be laid on the fundamental problems of power resources, their development and use as a means of promoting the social and economic interests of the nation.

The preliminary program, outlining the scope and subjects to be covered but not the authors, has been published in four languages—English, French, German and Spanish. The papers will also be published in these languages.

As a substitute for the technical program a number of tours are being arranged to include outstanding installations of steam power, coal production and processing, oil refining and hydrogenation and hydroelectric developments. Visits will also be made to several large research laboratories. In conjunction with these trips informal round-table technical conferences are to be held at centers visited. Some of these trips will precede and others follow the conference in Washington.

The American National Committee has its headquaters in the Interior Building at Washington with O. C. Merrill, Director, in immediate charge of the organization and planning.

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Progress in Power

For some years past it has been customary for the several professional divisions of the A.S.M.E. to prepare progress reports on trends and accomplishments in the respective fields for presentation at the Annual Meeting in December. At the last meeting the report covering fuels, steam power, oil and gas power, and hydraulics, was presented by Professor C. E. Lucke of Columbia University and approached the subject from a rather distinctive angle. The full report is scheduled to appear in the February issue of *Mechanical Engineering*, but the following is an excerpt from the introduction.

THE true objective of progress in power is reduction of power cost, or improvement of suitability of generating plant for a special case or class of drive in any of the divisions of transportation or stationary service. Whatever has reduced power cost is thus proved to be an item or a means of power progress, and the same is true of anything that improves the suitability of equipment for a given service without increase of power cost beyond what can be borne. There can be no absolute measure of progress. Comparison of new with old on a competitive basis as to power cost and service suitability must determine when progress has been made. Those new engineering ideas or inventions, newly developed processes or equipment that are the result of a power progress motive, may or may not finally be entitled to such a claim. The conclusion must be deferred until after acceptance and use in competition with alternates. Progress certainly is not measured by novelty alone, whether engineering progress or power progress.

While conclusion must thus be deferred, there is value in a weighing of possibilities of engineering improvements as prospective means of making progress in power. This sort of thing is, moreover, quite necessary in the formulation of plans, programs and policies by engineers and organization executives concerned with any phase of power activity. Usually each one prefers his own opinion, and there is just as much competition between opinions as there is between alternate power generating systems and equipment.

Interpretation of facts to discover trends may be as important as the facts themselves, and it may indicate directions worthy of effort in the interest of progress, or, negatively, directions in which no effort seems worth while. Identification of trends really amounts to forecasting, dangerous but necessary, and it requires information on both tangible things and intangibles, the interpretation of which may lead to conclusions as to what should or should not be done with no specific evidence in that direction.

Three Types of Persons Concerned with Power Progress

There are three major types of parties concerned with power progress and each may properly set up different

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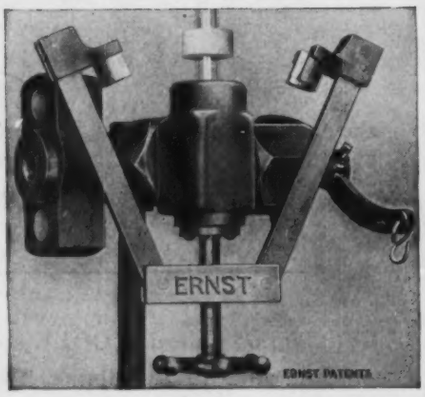

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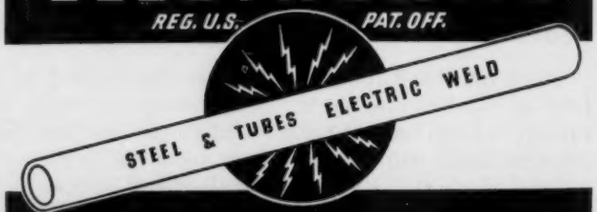

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standards of judgment as to advances in its own zone of activity; (1) The power user and engineer of utilization or application of power; (2) the manufacturer of power generating equipment, and the designing engineer; and (3) the operator of power equipment and the operating engineer, independent of power use and of creation of means of power generation. The power user must be put in first place as the supreme court of judgment of competitive cost of power and of the suitability for his purpose of alternative means of generation, the supplying of which is the business of the other two groups.

The other two groups whose attention is necessarily concentrated on alternative means of generation must deal with sources of energy, efficiency of transformation, apparatus design and performance limits, but here novelties that are engineering improvements worthy of notice as items of engineering development may not be classed as real contributions to power progress until accepted by the first group, the user, as competitively better for his purposes than what had been available. Admiration for the ingenuity of the designer or inventor, for the profundity of knowledge of the research engineer or the resourcefulness of the solver of technological problems is apt to obscure the fact that the result is acceptable and salable to the user only when the new generating equipment produces cheaper power, or is more suitable for his needs.

While the user group is the one that must pass judgment on progress as to competitively acceptable improvements in generating equipment, the exercise of this judgment by one class of user, has been too often limited by disinclination to change an old practice, thus minimizing the prospective value of new means perhaps developed and in successful use by another class of user and so close the door to possible progress arising from the efforts of the other two groups concerned with the improvements of means. Open-mindedness on the part of user groups is a prerequisite essential to all power progress.

There is evidence of progress by all three groups, both tangible and intangible, the tangible being concerned with creation of improvements in new equipment, in operation of equipment, and in user acceptances, especially as between different groups of users, such, for example, as stationary in relation to transportation classes of users. Some observations and comments on progress are given in the report in four classes, the first being general as to systems of generation, and the last three based on the steam, internal-combustion and hydro systems, respectively. Some are items of power progress, others of engineering development, and each reader may judge for himself the relation of the latter to the former.

Intangible progress is evidenced by rapidly growing acceptance by all three groups of research, properly defined and properly directed, as a necessary activity instead of something to be suppressed as a nonproductive expense. This was mentioned last year with reference to means of generation, but it is being extended to the user problem of selection or adoption of alternate driving equipment. The key to this problem is rationalism of analysis of relations between cause and effect, as distinguished from empiricism or tradition, and rational analysis has never been so actively accelerating in so many directions as now.

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ENGINEERING BOOKS

1—Reducing Industrial Power Costs

By David Moffat Myers

378 pages Price $4.00

This book deals with the generation and utilization of steam and power from the economic angle rather than from that of design. It is not written for technically trained power engineers but rather for executives and managers, with or without some engineering background, who have supervisory power responsibilities and may at times be faced with the necessity of making decisions in such matters. Hence, the text is essentially non-technical and of an educational or informative character.

Chapters on the selection of fuel, making or buying power, control of efficiency, steam plant losses and savings, the human factor, etc., are typical of the economic treatment. Other chapters on the boiler plant, steam pressures, fuel burning equipment, prime movers, boiler water treatment, heating systems, etc., serve to give the reader a rather general idea of practice.

The "Power-Loss Chart" provides a most convenient method of checking and correcting losses that may be going on in an industrial power plant, and a concluding chapter on "Power Facts and Data" provides a useful reference for comparison.

2—The Chemical Formulary Volumes I and II

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Chemists, engineers, technical workers, purchasing agents, executives, instructors, etc., will find this condensed collection of valuable, timely and practical formulas for making thousands of products in all fields of industry of great assistance. It bridges the gap between theory and practice. More than two hundred industrial chemists, professors and technicians in many branches of industry have cooperated to make this the most modern work of its kind available.

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3—Combustion in the Power Plant (A Coal Burner's Manual)

By T. A. Marsh

255 pages Price $2.00

The author's discussion of coals and combustion is simple and understandable. His consideration of equipment—stokers, boilers, furnaces, fans and auxiliaries—is thoroughly practical. He tells how to select a stoker for the best available coal; how to design furnaces and arches; how to analyze draft problems and design chimneys, gas flues and boiler passes; how to purchase coal and calculate steam costs. He gives to every phase of his subject a practical interpretation that makes this book of exceptional value to men actually identified with steam plant design and operation.

4—Coal through the Ages

By Howard N. Eavenson

123 pages Price $1.50

This little volume is a compilation of addresses made by the author during presidential visits to various local sections of the A.I.M.E. during 1934 and is published for the Seeley W. Mudd Fund by the society. The text traces in a most fascinating manner the relation of mineral fuels to advances in civilization; it describes the early mining methods and their subsequent development, and discusses coal as a pertinent factor in our national economy. Well executed pen sketches add to the attractiveness of the book.

5—Dust

By S. Cyril Blacktin

296 pages Price $6.50

Dr. Blacktin has sought to supply the need for a comprehensive monograph on dust, and has written an exceedingly useful treatise. The scope includes every aspect of dust, and his book will be valued for the bibliography alone, which lists more than 500 references. Among other subects the author deals with dust storms, explosive dusts and diseases caused by dusts.

The consideration of dust and its effects is required in widely diversified branches of science and technology, including geology, botany, agriculture, theoretical research, smelting and refining, air conditioning and many processing industries.

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6—Fans

By Theodore Baumeister, Jr.

241 pages Price $3.50

The author, who for several years has conducted a course in air and gas machinery at Columbia University, has prepared this book not only as a reference in teaching but also as a practical treatise for those who from time to time may be called upon to select fans for specific purposes.

For this reason the text has been arranged logically by first describing the principal types and commercial makes of fans and blowers; then discussing the characteristics of these types, in which connection terms and laws are explained; and then a chapter on selection, which includes numerous tables. "Theory and Design," "Fluid Flow" and "Fan Testing" are the titles of subsequent chapters for those who may have occasion to go into this phase of the subject.

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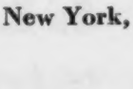

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Boiler, Stoker, Pulverized Fuel

as reported by equipment manufacturers of the Department of Commerce, Bureau of the Census

Boiler Sales

Orders for 81 water-tube and h.r.t. boilers were placed in November

	Number	Square Feet
November, 1935	81	224,863
November, 1934	55	138,745
January to November (inclusive, 1935)	929	3,054,951
Same period, 1934	802	2,307,647

NEW ORDERS, BY KIND, PLACED IN NOVEMBER 1934–1935

Kind	November 1935 Number	November 1935 Square Feet	November 1934 Number	November 1934 Square Feet
Stationary:				
Water tube	44	178,180	32	115,692
Horizontal return tubular	37	46,683	23	23,053
	81	224,863	55	138,745

Mechanical Stoker Sales

Orders for 213 stokers Class 4* totaling 34,849 hp were reported in November by 68 manufacturers

	Installed under: Fire-tube Boilers No.	Fire-tube Boilers Horsepower	Water-tube Boilers No.	Water-tube Boilers Horsepower
November 1935	170	21,463	43	13,386
November 1934	180	22,146	25	17,621
January to November (inclusive, 1935)	1,595	207,548	522	193,502
Same period, 1934	1,517	189,246	446	182,700

* Capacity over 300 lb of coal per hr.

Pulverized Fuel Equipment Sales

Orders for 15 pulverizers with a total capacity of 126,035 lb per hr were placed in November

STORAGE SYSTEM

	Pulverizers: Total number	No. for new boilers, furnaces and kilns	No. for existing boilers	Total capacity lb coal per hour for contract	Water-tube Boilers: Number	Total sq ft steam-generating surface	Total lb steam per hour equivalent
November 1935	..	..	..	.....	..	.....	.....
November 1934	..	..	..	.....	..	.....	.....
January to November (inclusive, 1935)	..	..	..	.....	..	.....	.....
Same period, 1934	2	1	1	46,000	*	*	*

DIRECT FIRED OR UNIT SYSTEM

	Pulverizers				Water-tube Boilers		
November 1935	13	11	2	124,650	10	87,403	988,900
November 1934	3	..	3	6,335	3	11,520	67,800
January to November (inclusive, 1935)	123	86	37	798,440	98	708,587	6,803,560
Same period, 1934	82	57	25	502,785	65	412,017	4,172,370
					Fire-tube Boilers		
November 1935	2	1	1	1,385	2	3,450	16,250
November 1934	2	..	2	1,535	2	3,000	15,600
January to November (inclusive, 1935)	7	1	6	6,685	7	12,830	68,750
Same period, 1934	12	3	9	11,765	13	15,486	111,900

* Data not available.

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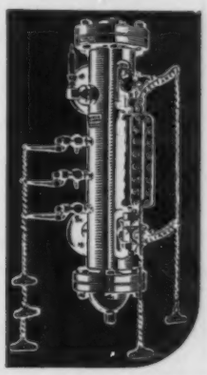

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Armstrong makes a complete line of traps for discharging air, or other non-condensible gas, from water or other liquids. Removal of water from compressed air and deaerating closed water systems are typical applications. Ask for a copy of the new Armstrong Air Trap Bulletin. It describes numerous interesting applications.

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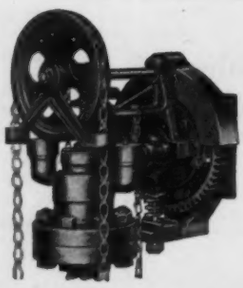

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